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DIESEL ENGINES

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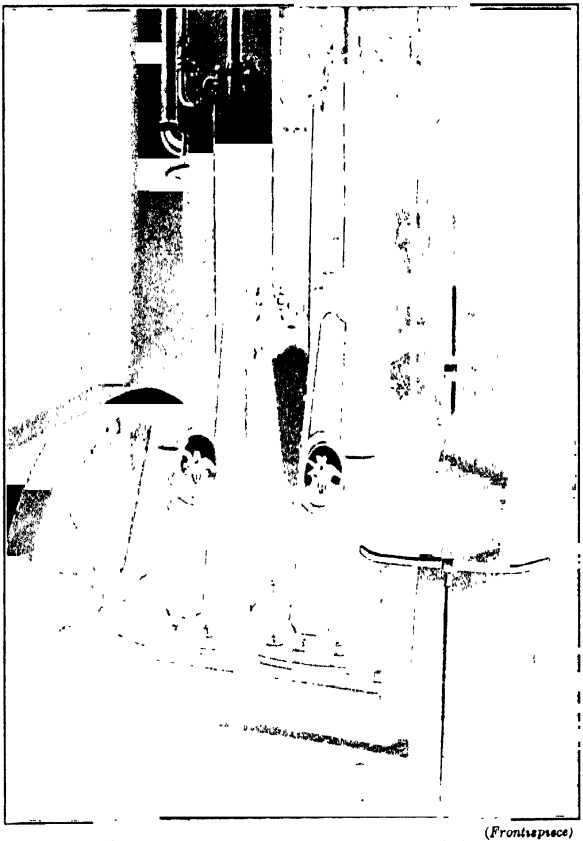

(*Frontispiece*)

First Diesel built in the United States—1898.

DIESEL ENGINES

BY

LACEY H. MORRISON

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First Edition

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PREFACE

The increase in the cost of power in coal burning power plants has caused many to give serious consideration to the Diesel engine. That this type of engine has an assured future is evidenced by increase in yearly sales. Several years ago the author ventured to assemble a considerable amount of data on the internal combustion engine which was published in book form under the title of "Oil Engines." At that time only a few firms were building Diesel engines and such material as was available was included in the discussion of all engines in general. At present there are more than twenty five concerns building or experimenting with the Diesel engine and the information has increased correspondingly.

Since the economics of this type of engine are as vital as the operating procedure, considerable space has been devoted to production costs, which should be of as much interest to managers of plants now using steam or purchased current as to oil engine plant owners.

The author wishes to make due acknowledgement of the courtesies extended him by the engine builders as well as to *Power* for permission to use some material which appeared in that publication. Thanks are due also to the author's wife, Emma Snow Morrison, to whose assistance and encouragement this volume owes its existence.

L. H. M.

PORT WASHINGTON, N. Y.
August, 1923.

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DIESEL ENGINES

CHAPTER I

INTRODUCTION

The Diesel-engine Industry in America

General.—To those not intimately acquainted with the Diesel-engine industry in the United States the statement that the total output of the several manufacturers since the first introduction into this country amounts to 500,000 hp. will be somewhat surprising. Particularly interesting is the development since the expiration of the original patents. The wonderful impetus given the industry by the engagement of a number of manufacturers in the building of Diesel engines serves to prove the statement that competition is always stimulating.

Installed Horsepower.—The increase in total horsepower sold per year, comparing the early years with 1921 or 1922, is quite comparable with the increase in the capacities of the individual units. In the frontispiece is shown the first engine constructed in the United States in 1898, being of 60 hp. One may gain an idea of the growth of the industry by comparing this engine with a modern Diesel of 2,000 hp. Likewise the yearly output has increased from 60 hp. in 1898 to 70,000 hp. in 1922, as will be noted from a study of the chart in Fig. 1, which first appeared in *Power* in an article by the author on the Diesel engine. In the preparation of this chart all the Diesel manufacturers were consulted. Since many of the older units have been re-erected in other than the original plants, it was necessary to trace a considerable number of these engines. For this reason a few minor errors exist in the totals for the first 10 years; these errors will not amount to over 2,000 hp. for the entire range of the chart. A few of the early engines have been discarded or are used as breakdown service.

Life of the Diesel Engine.—One of the interesting facts to be gained from this chart is the continued service obtained from practically all the engines constructed. Many engineers hold decided views against the oil engine, basing their prejudice on the statement that the Diesel, especially of the early design, has been a failure. To the contrary, no other prime mover can show such a record of continuity of service. If, out of a total of 500,000 hp. less than 8,000 hp. are no longer in service, the impartial investigator must decide that its life is fully equal to that of a steam unit. It is well to add here that of the total horsepower of discarded engines some 2,000 hp. were installed in a plant that was destroyed by fire from causes other than those arising primarily from the engines.

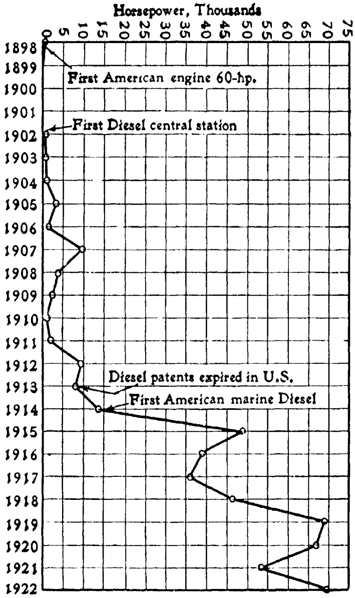

FIG. 1.—Growth of Diesel industry in the United States.

Industries Using Diesels.—The application of Diesel power is not confined to narrow limits, the engine being employed in a large variety of industries. In Fig. 2 are shown the total Diesel installations as reported Feb. 1, 1922, separated into the classifications of the industries served and the percentage of the total number of Diesels so employed. The 70,000 hp. sold in 1922 are divided in approximately the same ratio. As might be expected from its very apparent adaptability to such work, the greatest volume of Diesel horsepower is employed in the central stations and on oil pipe-lines. Since the business of the oil pipe-lines is to transport petroleum, the oil engine is especially suited for such pumping stations. As new oil fields are opened up the demand for this engine increases. With the developments in Canadian and South American fields a large number of American Diesels will be used since the American manufacturer has solved the problems of oil-line drive more successfully than have the foreign builders.

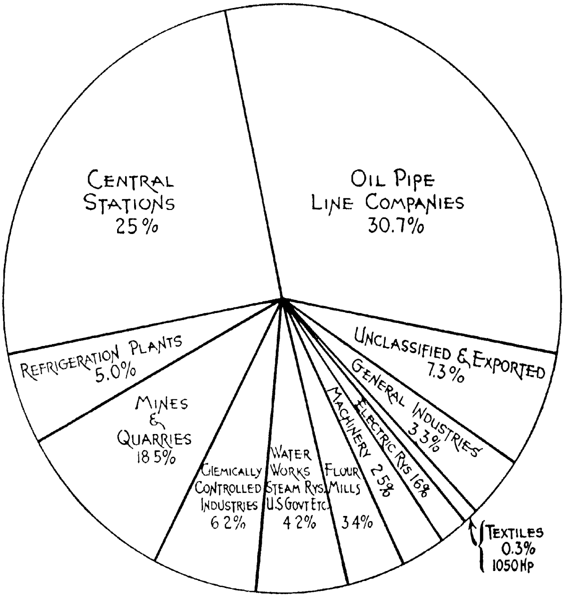

Fig. 2 —Industries using Diesels.

The Diesel has found a general acceptance in central stations of capacities under 2,000 kw. In these small- and medium-powered plants it is impossible to install an efficient steam plant with all the necessary auxiliaries. As a result, the Diesel has been largely sold in such plants. That the service has proved satisfactory is attested by many repeat orders.

The operating cost per kilowatt is well under usual steam-plant records and compares quite favorably with the cost of the largest central stations, as will be pointed out in Chapter V. The overhead charges, in view of the present costs of steam-plant equipment, are not excessive, as will be outlined in a number of Diesel central-station plant-cost records appearing in Chapter V. Since the Diesel is a self-contained unit, corporations controlling a number of central stations have found that it can be installed in one locality when the power demand is not high, and afterward when the load conditions justify a turbine plant, the Diesel can be moved to another station with practically no depreciation. Of the industries using Diesels it appears that while oil lines have purchased a greater yearly horsepower of Diesels, the growth of the Diesel central stations has been the most consistent.

The more general adoption of the Diesel to refrigeration plants is to be expected since the fuel cost per ton of ice with the Diesel is less than one-quarter the cost with the typical steam plant. The increased use of raw-water plants makes the oil engine the most economical prime mover when fuel-oil prices are reasonable, the fuel costs being as low as 15 cents per ton of ice. A study of Fig. 2 reveals that all the larger industries have to some extent made use of the Diesel; it is generally agreed that in every line of manufacture the oil engine is a competitor of other prime movers save in those cases where a large amount of exhaust steam is needed.

American engineers have not realized upon the heat in the Diesel exhaust as have foreign engineers. Since repeated experiments have proved that 15 per cent of the heat in the fuel can be recovered in the exhaust together with 12½ per cent from bypassing a part of the cooling water, it is apparent that more study should be given to this point. This is dealt with to some extent in the chapter on Cooling Systems.

The territory covered by the United States is so vast that fuel-oil prices vary over a wide range, dependent on the locality. Along the Atlantic seaboard the Eastern coals in the past have been cheap when compared with current fuel-oil prices. During the past 4 or 5 years the price of oil has been much lower than coal on the basis of "per million B.t.u." This tends to encourage a now general use of the Diesel in these states, and many plants are found to be operating profitably. In the Middle West, although coal is lower in cost than in the East, fuel-oil is obtainable at prices that place the steam plant at a decided disadvantage.

In the Eastern States the density of population and of industry is such that the central stations are of large capacities. The isolated plant, even when Diesel-driven is sometimes at a disad-

vantage. Notwithstanding these adverse conditions, a considerable capacity in Diesel engines is found in these states and sales are increasing. In the Southwest the industries are not so congested and the central stations are of medium capacity. These

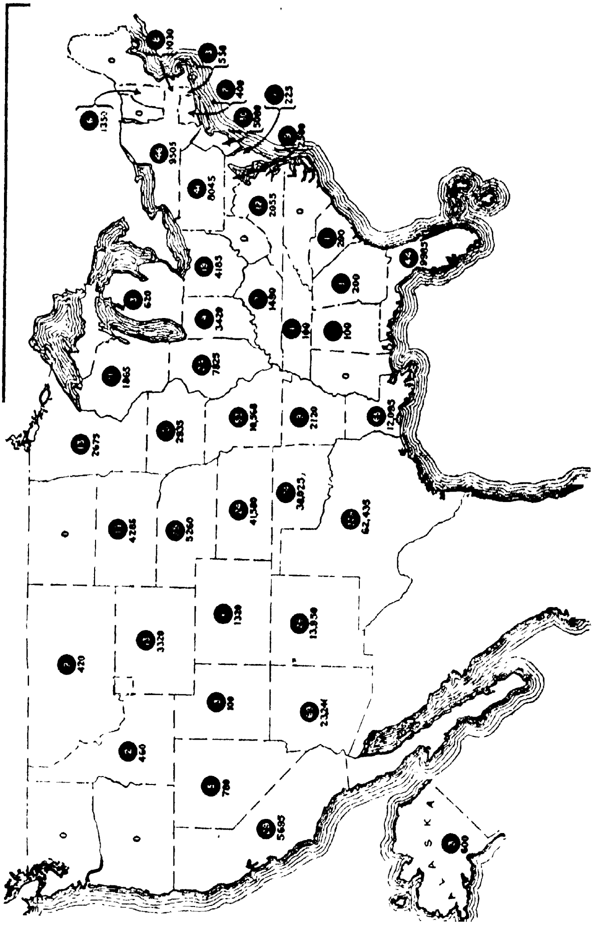

FIG. 3.—Diesel installations in 1921.

factors, together with attractive oil prices, make the Diesel engine especially suitable for the Southwest and Central States. The map in Fig. 3 shows the number and total horsepower of the Diesel installations in each state on Feb. 1, 1921, when the author made a very comprehensive survey of the United States. Five states of the Central West—Missouri, Kansas, Oklahoma, Texas, and Louisiana—contain one-half the Diesel horsepower of the United States.

As might be expected, one reason for the large number of Diesel installations in the Central West, other than favorable fuel-oil prices, is the fact that in these states are found a large number of oil pipe-line pumping stations. In the North Central States the Diesels are used almost exclusively in the central stations. A large percentage of the central stations in the Southwest are Diesel-engined. In the Far West the mines use most of the engines listed, although a number of small engines are found in irrigation plants. The development of irrigation projects in the Southwest will lead to a more general use of the Diesel. Large areas in Texas, New Mexico and Arizona will ultimately be put under irrigation, and pumping equipment will be necessary.

Marine Diesel.—The marine Diesel industry is still in its infancy; that the marine Diesel horsepower has almost reached 100,000, including those on order Dec. 1, 1922, is commendable. While the first marine Diesel was installed in 1914 and the largest units in America were built in 1915, there was no considerable activity in this field until 1918. The horsepower given above includes all save those in submarines; in this field there are approximately 140,000 hp.

There is every reason to expect a consistent demand for marine engines. In medium- and small-tonnage vessels they find an especially suitable field. In tank service no other form of power is so efficient. On the other hand, a large volume of marine engines, either of steam or oil, cannot be expected for several years. The world's demand for tonnage, save vessels for special requirements, has been fairly satisfied. There was less tonnage launched in 1920 than in 1919, and in 1922 a still lower amount. In addition, marine architects and shipbuilders have not been entirely converted to the Diesel drive. There is a strong undercurrent, however, among shipping circles that the Diesel is the solution of the high cost of operating U. S. vessels.

It is a matter of regret that the U. S. Shipping Board has failed to give proper encouragement to the Diesel builders.

Export Field.—The export field offers attractions to the American Diesel manufacturer. During the life of the original Diesel license it was not possible to export American Diesels; consequently, the export period started in 1914. Since that time 106 engines, totaling 23,470 hp., have been sold to various countries. This does not include, however, 900 hp. which were originally in operation in the United States and subsequently shipped to the West Indies, South America and Canada.

In a number of countries—the West Indies, South America and the Orient—the scarcity of coal has created a demand for the Diesel engine. European manufacturers have had representatives in these countries for years; as a result the major part of the demand has been filled by foreign firms.

Small Diesels.—Mention should be made of the activity in the building of Diesels of almost miniature capacities. Several concerns are manufacturing engines of less than 100 hp. One builder has devoted almost his entire attention to these small units and has sold a large number of engines of 75 hp. and under. This can be taken as indicative of the progress made toward the simplification of the Diesel so that it can be handled by the average mechanic after some shop instruction.

The advantage of a Diesel engine for small horsepower is its marked economy over the use of a steam plant. The gain in economy is due to the fact that as steam equipment becomes smaller, its fuel consumption per horsepower increases. It is also due to the fact that the small steam plant considers any steam unit good enough, which leads to great waste in fuel consumption. A Diesel engine, however, whether large or small, has approximately the same fuel consumption. This is an advantage which is being appreciated more and more and which makes a Diesel engine in smaller units very attractive. It also leads to the adoption of several units in one power plant in preference to one, making it possible to operate a Diesel engine at more nearly its economical load. By the use of a number of small units, the cost of the spare unit is kept down, as well as is the maintenance charge.

Large Diesel Engines.—Probably the most striking advance made of late years in Diesel-engine design has been the increase in engine horsepower. In the United States the tendency until

2 or 3 years ago was to hold the cylinder size below 150 hp. The manufacturers were content to build and market engines of 500 hp. and under. In fact the average engine rating, considering all the Diesels manufactured in this country, since 1898, is slightly under 275 hp. Since the Diesel demand came from small central stations, oil pipe line and like plants, low-powered engines were logical. The oil pipe lines absorbed more than 30 per cent of the engine output; these stations must have duplicate units, and 500 hp. is about the maximum size that can be used, although during 1923 engines of 750 hp. were installed. The small central stations, where efficient steam operation was impossible, required well under 500 hp. The engine builders largerly influenced this demand for only small engines, for it is only of recent date that the one time perplexing problem of crankshaft failure and cylinder-head fracture have been solved.

While the European builders have been turning out engines of large size for several years, the American manufacturers have recently shown a marked tendency toward the designing of large units; at the present time the maximum output of any American engine is 2,700 b.hp. While this Bethlehem engine is the largest in point of delivered horsepower, it has six cylinders and on the basis of individual cylinder capacity is now overshadowed by the largest Nordberg Diesel. The latter engine is of the two-stroke-cycle type and has four cylinders of 500 b.hp each. The design is such that two more cylinders can be conveniently added, giving a 3,000 hp. unit.

There is every reason for this increase in engine horsepower. Many plants of 3,000 to 5,000 hp. now operating with steam in unfavorable localities have found the power costs mounting to alarming figures. While fuel and labor costs may show some decrease in the future, they will never return to the pre-war level. It is impossible to consider replacing such equipment by ten or a dozen small Diesels. The maintenance costs, which are more or less in direct ratio to the number of vital working parts, would be prohibitive, while the labor would also be high since it is in ratio with the number of units installed. The production of these larger Diesels enables plants above 2,000 hp. to install oil engines. The cost per horsepower of the large units is well under the costs of small engines. The repair charge should be less, since better workmanship is possible and the character of operators can be higher.

CHAPTER II

INTERNAL COMBUSTION ENGINES

HISTORICAL

General.—While a complete history of the development of the internal combustion engine is of interest, such a treatment is beyond the scope of this volume. In order to obtain a comprehensive appreciation of the modern oil engine a brief statement of the more important steps in internal combustion-engine design is at least advisable, if not absolutely necessary.

Although for decades the internal combustion engine was neglected while the attention of the engineer was centered on the steam engine, still the former had been built and actually used long before Watts constructed his atmospheric steam engine. Because of defects of operation in the internal combustion engine that seemed impossible of solution, the steam engine took precedence, and it was only during the latter part of the nineteenth century that the gas engine began to come into commercial favor.

Huyghens.—While several scientists had suggested the use of gunpowder in a closed vessel as a source of power, Huyghens appears to have been the first who actually built an internal combustion engine (1680). His design embodied an open-ended cylinder with a piston. The powder was exploded when the piston was at the bottom of the cylinder: the explosion forced the piston upward. The gases were expelled through leather valves, and the cooling of the gases that remained in the cylinder created a vacuum. The atmospheric pressure, acting on the piston, forced it downward into the cylinder, doing work by its movement. Many structural defects caused the abandonment of this design.

Barber's Producer Gas Engine.—The internal combustion engine was practically ignored until 1791 when an Englishman, John Barber, built an engine which made use of gas distilled from coal. The essential features of Barber's engine were the use of a mixing chamber wherein the air and the producer gas were mixed and ignited, and the employment of a paddle-wheel against the blades of which the gases, issuing from the chamber, impinged. This machine was in reality a gas turbine rather than a gas engine, and several modern engineers have worked on the design of a gas turbine along the same lines.

Lenoir.—Although a number of experimenters interested themselves in the problem of building an internal combustion engine, Lenoir was the first to manufacture a marketable engine in 1860.

The Lenoir engine, Fig. 4, was double acting; the valves were of the flat-ported type, somewhat along steam-engine practice, being operated by eccentrics. The gas and air were mixed in the inlet valve and passed into the cylinder under the suction effect of the retreating piston. When the piston was almost halfway in its stroke, the charge was ignited. The presure then rose to the vicinity of 75 lb. The piston continued to the end of its stroke, being acted upon by the gas pressure. The exhaust valve then opened, and the burnt products were pushed out by the piston on the return stroke. Hundreds of these engines were placed in commercial use, but the gas consumption was high, the efficiency being but little above 4 per cent.

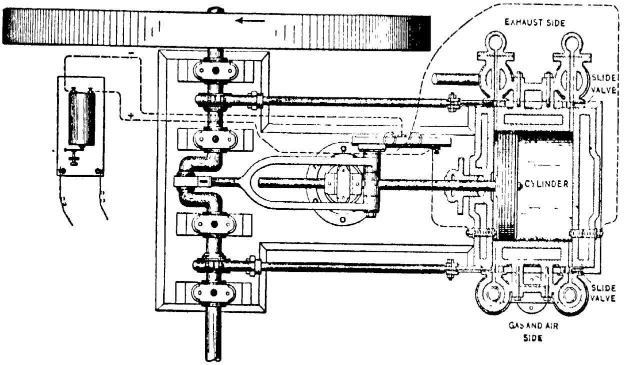

Fig. 4.—Lenoir's gas engine.

Otto's Engine.—About this time, 1866, Otto, in Germany, designed an engine with a free piston. In this engine, which was built vertically with the cylinder below the crankshaft, the piston was forced upward by the explosion. The weight of the piston then caused it to descend into the cylinder, assisted by the vacuum formed by the cooling of the gases. In descending, the piston engaged a rack which turned the crankshaft through a clutch. The inertia of the flywheel started the piston on the upward stroke, and the vacuum formed drew in a gas charge. This charge was exploded when the piston was partly advanced on the upstroke. It is apparent that this embodied the same principles as did Lenoir's engine.

Beau de Rochas.—In 1862 Beau de Rochas, a Frenchman, proposed the cycle that is at present used by the majority of gas engines. His proposal, although he never actually built an engine, was as follows:

The essential factors in obtaining high efficiency are: first, the highest possible compression pressure at the moment of ignition; second, the greatest possible expansion of the gases after combustion.

To achieve the desired efficiency in an engine, de Rochas proposed that an engine should operate on the four-stroke-cycle principle, having the following events:

Stroke 1. Draw in the air and gas charge.
Stroke 2. Compression of the charge.
Stroke 3. Ignition and expansion of the charge.
Stroke 4. Discharge of the burnt gases.

It is to be observed that these events are those occurring in all present-day four-stroke-cycle gas or gasolene engines.

Otto's Silent Engine.—As stated heretofore, Otto has obtained a patent on a free piston engine in 1866. In company with Langen, Otto formed an engine-building organization—the Gas-Motoren Fabrik Deutz—which is still in existence. Objectionable features of the Otto and Langen engine caused them to adopt the design proposed by Beau de Rochas. This engine, which was known as the "Otto Silent," created a furor at the Paris Exhibition in 1878. Figure 5 shows a view of one of these early Otto engines, while Fig. 6 is a section through the intake valve. This valve has two passages in it. The port *M* is the air passage, which, on the suction stroke of the engine, is in line with *m*, the air-suction pipe; Q is, at this time, in contact with *a;* and *n* communicates with *L*, the gas passage. This allows the air and gas to enter the cylinder through *a*. As the valve continues to move to the left on the compression stroke of the piston, *N* receives a small charge of gas from the gas line *d*. As the valve moves to the right on its return stroke, the cavity *N* moves past *B*, which carries an open flame, and the gas in *N* ignites. When this passage moves past the passage *a*, the flame in *N* ignites the gas in the cylinder. The exhaust valve, not

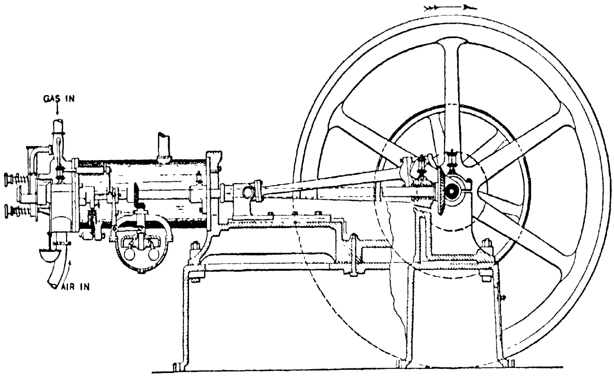

FIG. 5.—Otto's silent engine.

shown, was of the poppet type. The disadvantage lay in the flat admission valve. To keep it against its seat, a strong spring pressure was necessary; this occasioned rapid wear. However,

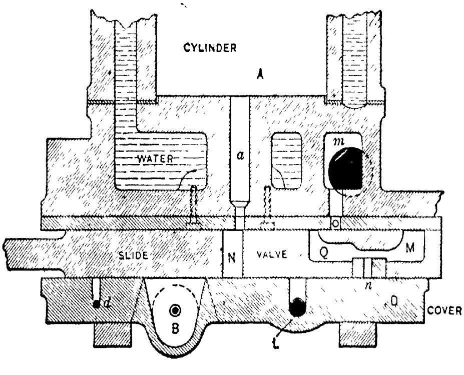

FIG. 6.—Cross-section of Otto's silent engine valve.

the engine was so superior to its competitors that thousands were sold all over Europe. It is to be noticed that the cycle used should have been termed the "Beau de Rochas cycle" instead of Otto, by which latter name this cycle is now universally designated.

This engine, modified to better meet existing conditions, is in daily use all over the world; thousands are sold each year; one American firm in 1921 marketed 60,000 Otto-cycle engines in sizes from 1½ to 10 hp., and there are scores of manufacturers who produce from 1,000 to 5,000 engines yearly; in 1921 the sales of such engines amounted to 533,006 hp.

Brayton's Constant Pressure Engine.—A few years previous to the design of the Otto silent engine, Geo. Brayton, a Philadelphian, secured patents on an engine that differed from all other internal combustion engines in that, instead of the charge burning at constant volume or instantaneously as in the Otto cycle, the fuel, either gas or liquid, was introduced into the cylinder in such a manner as to cause it to burn at constant pressure. The design of this engine embodied the use of an engine cylinder and of a separate charging or compressing cylinder. The gas and air were introduced into this charging cylinder and compressed. At the beginning of the power stroke in the engine cylinder the mixture passed into the cylinder through the intake device, which was in the form of a gauze screen. A pilot light ignited the charge as it blew into the cylinder until cut-off took place at about 10 per cent of the engine stroke. The flow of the mixture was such that the pressure in the power cylinder did not increase during combustion. After the flow of fuel ceased, the burnt gases expanded as in any engine. This Brayton cycle had much to commend it. It was the most efficient of all cycles operating between the same temperature limits. Its drawback was that, for equal power, it required a much larger cylinder volume than did the Otto cycle. The serious objection, both from a manufacturing and operating standpoint, was the excessive size and weight of the engine as well as the complicated mechanism. This cycle was early abandoned in favor of the more simple Otto cycle.

Clerk's Two-cycle Engine.—This engine was the design of Dungan Clerk, and, in place of being four-stroke cycle, the engine was of the two-stroke-cycle type. The design embodied the use of a charging cylinder into which the air and gas was introduced and compressed. The engine cylinder used no exhaust valves but, instead, made use of ports about the cylinder. After the power stroke was almost completed, the piston uncovered these ports, allowing the exhaust gases to blow out. At the same time the changing cylinder forced a charge of air and gas into the engine cylinder. This charge assisted in freeing the cylinder of the remaining consumed gases and was compressed by the piston on its return stroke. At the proper time a gas flame, and in some engines a hot bolt on the piston, ignited the charge. This cycle was not so favorably received since it was not as simple as the Otto engine. Subsequently, many engines were built using this two-cycle principle but compressing the gas and air in the crankcase in preference to the separate charging cylinder. The Clerk engine was the forerunner of the modern two-cycle engine.

Hornsby-Ackroyd Engine.—This engine, which was the pioneer of all the Hot-bulb or "Surface-ignition" engines, was an English product, being the invention of Charles Ackroyd Stuart, at present a resident of Australia. Hornsby and Co. took up the manufacture of the engine and called it the "Hornsby-Ackroyd." The De La Vergne Machine Co. acquired the American license, marketing the engine under the name "H. A. Type." In time the American rights were purchased by this company, and the former arrangement of turning over all data to the English company was no longer in force. As a natural consequence, the American and English developments were along totally dissimilar lines.

This engine was of the four-stroke-cycle type, the valves being placed in a valve chamber at one side of the cylinder. The cylinder head included a vaporizer or uncooled bulb. The oil was injected into the bulb and, owing to the intense heat contained in the walls, was vaporized. At the end of the compression stroke the heat of the bulb, increased by the heat caused by the compression in the cylinder, was high enough to ignite the fuel. The engine was an explosive or constant-volume engine and followed the Otto cycle in its action. Figure 7 shows the vaporizer of this engine. It will be seen that the opening into the cylinder is constricted. In this way the fresh air in the cylinder has no chance to mix with the vaporized oil until the piston forces the air into the vaporizer.

Since the engine followed the four-stroke-cycle principle, the air charge was drawn into and compressed in the engine cylinder. The compression pressure seldom exceeded 60 lb. per square inch. While this was as high a pressure as the first engine could use due to preignition troubles, later modifications of this type have the compression as high as 250 lb. per square inch. The variation in compression depends upon the time of depositing the fuel charge in the vaporizer. In the early models the oil entered the cylinder very early in the compression stroke, while the later designs have the oil injection occurring during the last 45 deg. before dead-center.

The hot-bulb or vaporizer engine came into general favor by reason of its ability to burn kerosene (termed paraffine in England) with success, and the simplicity of design, which made it far superior to the ordinary four-stroke-cycle gasolene or carburetor engine.

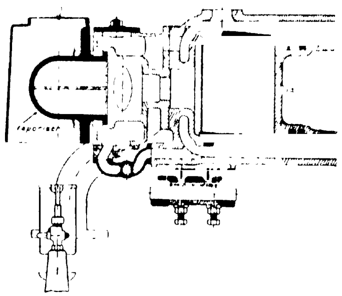

FIG 7.—Ackroyd hot-bulb engine.

The trend during the past 15 years is toward a two-stroke-cycle engine using the crankcase on front of the cylinder as a scavenging air compressor. This present-day semi-Diesel engine is, in respect to the method of fuel ignition and combustion, the original Hornsby-Ackroyd, the chief departure from this original design is the change from four-stroke cycle to two-stroke cycle. This change lowers the manufacturing costs and in some respects simplifies the operating details. In the number of builders and in the total horsepower capacity of these two-cycle hot-bulb engines the United States easily ranks first. In England and on the Continent more attention was given the carburetor engine, and it was as late as 1910 before the hot-bulb machine came into favor. At the present time they are finding a field of usefulness in stationary work and in small vessels; such as fishing boats, tugs and cruising yachts.

The Diesel Engine.—Dr. Rudolf Diesel, a German engineer, designed an engine to operate on the Carnot cycle, which calls for the heat to be added at the highest temperature and thrown away (exhausted) at the lowest temperature. In this engine the air charge on the compression stroke was to be compressed to a pressure such that the resulting temperature was above the ignition temperature of the fuel. The fuel was to be introduced into the cylinder at a rate such as to cause the work performed on the moving piston to absorb enough of the heat of combustion to keep the temperature constant. This would cause the cylinder temperature to remain constant during the period of fuel introduction extending over some 10 per cent of the piston stroke. After "cutoff" the gases were to expand adiabatically (without absorbing or losing any heat save that consumed in doing work). The compression of the air charge was to be at constant temperature (*isothermally*) for part of the stroke, followed by adiabatic compression (no heat lost). The engine was originally designed without water-jacket cooling since there was to be as little heat loss as possible.

The original patent, dated 1892, outlined an engine wherein the fuel used was to be pulverized coal or coal dust. This coal was to be stored in a hopper immediately above the engine-cylinder head. Between the hopper and the cylinder was interposed a rotary valve having a cavity or pocket. The valve received a charge of coal dust and in rotating in its seat came in communication with cylinder. The coal dust then dropped into the combustion space as the piston reached the end of the compression stroke. The fuel charge was to be varied to suit load conditions through the governor control of the valve movement.

Another interesting feature was the proposed introduction of a water charge at the beginning of the compression stroke for the purpose of securing isothermal compression during part of the stroke.

A second claim embodied in the same patent covered the use of liquid fuels with the employment of a spraying valve but without the air injection feature. The engine was to be started by some explosive agent in the cylinder for the initial stroke. The expansion was to be carried to such a point that the exhaust gases were to be cold enough to be used as the cooling medium in the cylinder jacket.

An engine was constructed along these lines but never turned over beyond the initial stroke, during which it was completely wrecked. At this late date no information seems available as to the cause of this wreck, but it is to be presumed that the starting charge of explosives was the destructive agent.

This disappointment caused the builders of this engine to abandon Dr. Diesel's original ideas, and a water-cooled engine was built in which the admission of heat was at constant pressure instead of at constant temperature as originally contemplated by Dr. Diesel. The first of these later engines actually ran but never was able to carry any load. Ultimately alterations were made which transformed the experiment into a practical machine.

The development of the Diesel proceeded slowly for some time. Various modifications were introduced in the engine which was built at the shops of the Maschinenfabrik of Augsburg; this concern in conjunction with Krupps bore the expense of the experiments carried on by Dr. Diesel and Ing. Vogel. It was not until 1897 that a commercial engine was produced, which was a single-cylinder 25-hp. engine of vertical design and using a crosshead. This was followed by the exhibition at Munich, 1898, of three Diesels by the M. A. N., Augsburg, by Krupps, and by the Deutz Co.

Starting with the modest-powered engine of 25 hp. it was but a question of a relatively few years before engines of a thousand horsepower were in commercial use. The engine was patented in practically every country, and for a few years all European manufacturers operated under a license; this, however, was discontinued around 1904, and but few manufacturers paid royalties. The popularity of the Diesel engine in Europe has been due, to a large extent, to the type of manufacturer building these engines and to the high cost of fuel. In Germany a number of the strongest steam-engine builders, having the facilities to do the extensive experimenting necessary, early took up this engine. In Switzerland and the Scandinavian countries the engine found early favor. The British Diesel Engine Co., even when the engine temporarily fell into disrepute in Germany, continued their labors, and much credit for the successful outcome of the Diesel, usually attributed to German firms, actually is due to the activities of the British manufacturers.

The Diesel was introduced into the United States by Adolphus Busch of St. Louis. The first American Diesel engine was completed in 1898, being of the two-cylinder vertical A-frame design, and developed 60 hp. A photograph of this engine appears as the frontispiece. The early history of the Diesel engine in America was one of disappointment, part of the adverse criticism of the engine that is encountered even now is the result of the policies of the early firms who attempted to follow gas-engine design of a character ill adapted to the conditions peculiar to Diesel-engine operation. During the past 10 years some twenty American engine builders have taken up the manufacture of the engine; the designs, in the main, follow European practice.

Diesel Marine Engine.—It is for marine work that the Diesel possesses many favorable characteristics. The first marine Diesel was constructed in 1903 by French engineers for use on a canal boat. This particular engine was single cylindered with opposed pistons and worked on the four-stroke cycle. While French firms produced a considerable horsepower of the marine engines, the greatest development occurred in Russia where the demand for river-boat engines stimulated the activities of the Diesel builders. Credit must also be given to Nobel Bros. of Petrograd for the production of the first Diesel for submarine service in 1908. This phase of Diesel manufacture has received great attention since 1914, and undoubtedly the greatest improvements in design have occurred on the submarine engine, especially as to large output per unit cylinder volume. During the World War the output of high-speed marine Diesels, both for surface and submarine service, was enormous. It has been stated by a high authority that 500,000 hp. of this type of engine was furnished to England, while the total submarine Diesel horsepower of United States is in the neighborhood of 150,000 hp.

While of late years engines have been built having a capacity of 2,000 hp. per cylinder, the American operator is actually concerned in engines ranging below 2,400 hp. per unit. As a consequence, attention for the most part will be centered on this class of engines.

The Ackroyd Cycle.—The engineer will observe that the engine now marketed as the Diesel does not follow Dr. Diesel's theories. Since Charles Ackroyd Stuart, long before Diesel's treatise, had patented an engine in which the air charge only was compressed in the cylinder, the fuel being introduced by a pump, gasified, and ignited by the heat contained in the air charge although assisted by a hot vaporizer, it would seem that the cycle could quite properly be termed the Ackroyd Cycle. True, Stuart employed a low compression pressure and depended upon the hot walls of the vaporizer to raise the temperature to the ignition point of the fuel, but he was the first to advocate the compression of pure air and the ignition of the fuel by the cylinder temperature without the aid of an electric spark. Having made his engine prior to Dr. Diesel's work, there are many who claim that the latter used the Ackroyd engine as the basis of the Rational Heat Motor.

CHAPTER III

THE DIESEL CYCLE

The Diesel Principle.—As has been outlined in Chapter II the chief point which distinguishes the Diesel from other forms of the internal-combustion engine is the compression of a charge of air until the resulting high temperature is sufficient to ignite the fuel without the aid of any external object. The fuel is forced into the cylinder by a blast of high-pressure air coming from an air compressor.

At the present time there are a number of oil engines manufactured both here and in Europe which, while using a fairly high compression, sufficient to secure ignition of the fuel, do not operate on the Diesel cycle but on what is a combination of combustion at constant volume (the Otto Cycle) and at constant pressure (the Diesel Cycle). These engines do not make use of an air blast as the means of oil injection, but depend, instead, on a direct pump action. For this reason, they are familiarly known as the "Solid-injection" oil engine and have also been called "Dual Cycle" engines on account of the constant volume and constant pressure combustion. This type of engine is treated in Chapter XX.

The features which set the true Diesel apart from other oil engines are: (*a*) Compression sufficient to produce auto-ignition of the fuel; (*b*) injection of the fuel by an air blast; (*c*) combustion with practically no change in pressure from the maximum compression pressure.

Cycle of Events.—Figures 8 *A* to *D* show the valve action and the conditions existing in the engine cylinder when the piston is at different points in the engine cycle. The pressure in the cylinder at any point in the piston stroke is indicated in Fig. 8 *F* while Fig. 8 *E* shows that part of the crank circle swept by the crank during any of the several events in the engine cycle.

Figure 8 *A* covers the suction, or admission stroke of the piston. The admission, or air-suction, valve at *J* has opened at the point *a* just before the piston has reached upper dead center. This valve remains open from the point *a* to the point

b during which time the piston moves downward, drawing in a cylinder charge of air, reverses and starts to move upward. This does not mean that the piston pushes part of the air charge back out through the admission valve. In operation the rush of the air through the air valve continues even after the piston starts upward. The momentum of the air causes it to "pile up" in the cylinder; in this way the amount of air entering the cylinder is actually greater than if the valve had closed at bottom dead center. This admission stroke is shown in Figs. 8 *E* and *F*; in the latter the indicator diagram shows this line as being slightly below the atmospheric pressure line *xy*. In Fig. 8 *B* the admission valve *J* has closed at *b* and the pure air charge

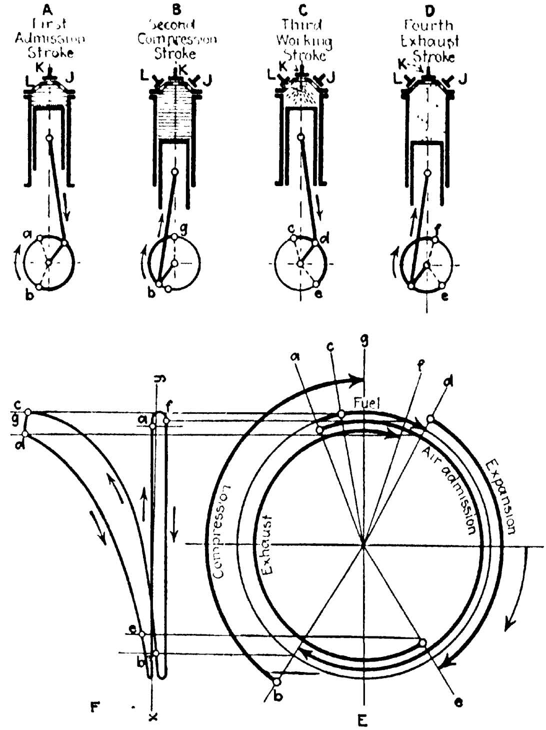

FIG. 8.—Cylinder events of a four-stroke-cycle engine.

is compressed by the piston up to the point *g*, which is top dead center. This process is covered by the compression line *bc* on the indicator diagram in Fig. 8 *F*. The clearance volume is very small, and the maximum or final compression pressure rises to some 450 to 550 lb. per square inch. The work done on the air charge in compression causes the temperature to rise to a final temperature of 900 to 1,400 deg. F., the air being in a red-hot condition.

At the point *c* in Fig. 8 *C* the injection valve *K* opens, and a charge of fuel is blown into the cylinder by means of a blast of high-pressure air. The fuel-injection valve or spray valve, as it is often called, is designed to cause the rate of flow through the valve to be regulated so that the entire oil charge is not injected instantaneously, but takes place while the engine crank turns through a considerable angle. In Fig. 8 *C* the injection of fuel starts when the crank is at *c* and ends when the crank is at *d*. In Fig. 8 *F* the line *cd* represents the fuel admission and combustion period, and the desired condition is attained when the line *cd* is practically horizontal, showing that the rate of heat addition is such that there is no increase in the cylinder pressure.

The injection and the combustion of the fuel ceasing at the point *d*, the piston continues to the end of its stroke under the influence of the expanding gases. Before the completion of the stroke the exhaust valve *L* opens when the crank is at *e*. This allows the gases to rush out through the exhaust passage. The exhaust valve continues to remain open until the piston again ascends to the top of the cylinder, expelling all the exhaust gases. This part of the cycle is shown in Figs. 8 *E* and 8 *D* as continuing from *e* to *f*. In Fig. 8 *F* this forms the exhaust *ef*. Before the exhaust valve *L* closes, the admission valve opens at *a*, allowing a fresh air charge to be inducted into the cylinder during the stroke shown in Fig. 8 *A*.

These events complete the cycle of the four-stroke-cycle Diesel. From a practical viewpoint the differences between the Diesel and the gas engine are that in the Diesel nothing but pure air is compressed in the cylinder and that the fuel is forced into the cylinder slowly, causing the combustion to be gradual; in the gas engine both the gaseous fuel and the air are compressed and the combustion takes the form of an explosion.

Summarizing, the Diesel four-stroke cycle consists of (1) a suction stroke during which a charge of pure air is drawn into the cylinder; (2) a compression stroke wherein this air is compressed by the piston to a maximum pressure; (3) a period of injection and combustion of a charge of fuel and the expansion of the gases, this being the working stroke of the piston, and (4) an exhaust stroke by means of which the burnt gases are expelled from the cylinder.

The Suction Stroke.—The air charge should be drawn into the cylinder at atmospheric pressure (zero gage, or 14.7 lb. absolute). Since the only reason air or any gas flows from one point to another is a difference in pressure, it follows that the pressure in the cylinder must be somewhat below that of the atmosphere if the air charge is to flow in and fill the cylinder. If there is a direct passage into the cylinder, this difference in pressure need be but a few ounces. Where a long air-suction line is used, such as installations with the air line leading from outside the building, the drop may easily reach 2 lb. This would give a pressure at the beginning of compression of 12.7 lb. in place of 14.7 lb.

On starting a cold engine the temperature of the air after reaching the cylinder, is at most no higher than that of the air of the building. The pressure drop, mentioned above, results in a temperature drop, and the cylinder air may be 10 deg. lower than the room temperature. In the cylinder into which the high-pressure air is turned in starting the engine, the expansion of this air will cause the cylinder to chill slightly. Upon shutting off the starting air, the suction air for several revolutions may be extremely cold from the absorption of its heat by the cold cylinder walls.

The Compression Stroke.—The purpose of the compression in any internal-combustion engine is to raise the pressure and the temperature of the air charge, or in case of a gas or gasoline engine the mixture of gas and air. It has been found that the efficiency depends upon the compression ratio and the temperature at the beginning and end of the compression stroke.

The compression pressure in the Diesel engine depends upon (1) the attainment of a satisfactory efficiency; (2) the securing of a temperature sufficient to produce self-ignition of the fuel and air charge, and (3) the mechanical construction of the machine.

Experience has demonstrated that the increase in thermal efficiency by increase of compression pressure above 500 lb. per square inch is too small to overcome the disadvantages from constructional and operating viewpoints. It is customary to carry a maximum compression of 480 to 520 lb., although in certain high-speed and two-stroke-cycle engines pressures up to 600 lb. are not unusual. Since the compression temperature must be above the ignition point of the fuel, there is a minimum pressure possible to employ in the true Diesel.

The temperature of the air charge at the end of the compression stroke must be high enough to ignite the fuel. If no heat was lost to the cylinder walls—that is, if compression was adiabatic—the relation of the pressures and temperature at the beginning and the end of the compression stroke would be expressed by the equation,

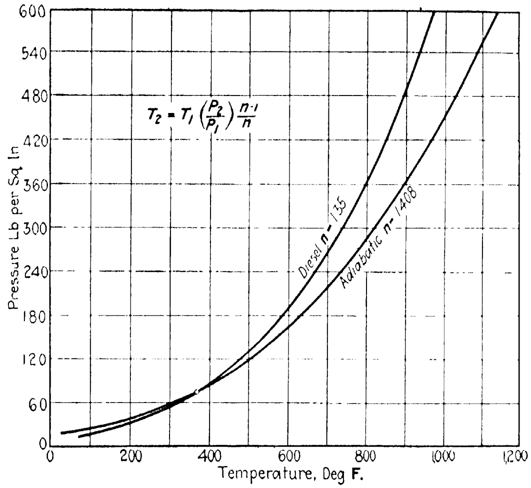

FIG. 9.—Relation of temperature and pressure during compression.

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

in which

T_1 = Temperature at the beginning of compression in deg. absolute (460 + F. reading);

T_2 = Final compression temperature;

P_2 = Final compression pressure, lb. absolute;

P_1 = Initial pressure;

n = 1.408.

This relation is shown in Fig. 9 by the curve marked adiabatic. There is a loss of heat during compression and expansion. By reason of this compression is not adiabatic, but is fairly represented by the expression with a value of n of 1.35. This relation is shown by the second curve in Fig. 9.

The relation of the volumes and temperatures are shown in Fig. 10 when V_1 = the volume before compression and V_2 = clearance volume.

The Combustion Period.—The fuel valve closes after the piston has traveled about 10 per cent of its stroke, the crank moving about 40 deg. of its cycle. The fuel and air mixture theoretically should be completely burnt by the end of the fuel injection, but in the actual engine the combustion continues far down the expansion line.

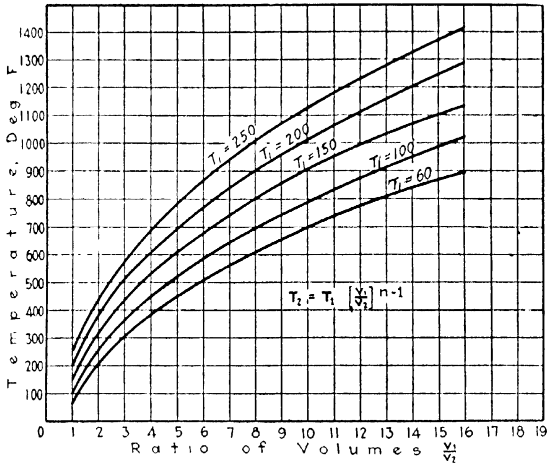

FIG. 10.—Volume temperature relation.

While the Diesel is spoken of as a constant-pressure engine—that is, the heat is added at constant pressure—it is seldom that an indicator diagram from an engine in every-day service shows a horizontal combustion line. A sloping line is customary and has been found to give a higher efficiency. With the ideal horizontal combustion line, while the pressure is constant, the temperature rises from a final compression temperature of, say, 1,200 deg. to some 2,700 to 3,500 deg. F.

At the beginning of this fuel-combustion period the opening of the fuel valve is followed by a slight time lag before the fuel enters the cylinder and ignites. This lag is due to the inertia of the fuel resting in the fuel valve, to the resistance of the atomizer disks, and to the slowness of ignition after the oil is in the cylinder. This delay in ignition depends upon the compression pressure and temperature and upon the pressure of the air spray.

The Expansion Stroke.—After the fuel valve closes theoretically, combustion should be at an end. The burnt gases now expand, forcing the piston outward and so increasing in volume and decreasing in pressure.

The Exhaust Period.—In the actual engine the exhaust valve opens before the end of the stroke, approximately 50 deg. ahead of crank dead center, or about 10 per cent of the stroke. The energy lost by early release is negligible, and if the release is delayed until dead center, the drop in pressure will be carried on during the part of the exhaust stroke, making the negative work of the exhaust stroke greater.

The temperature of the gases, when first issuing through the exhaust valve, is around 1,000 deg. F. This temperature does not exist beyond the valve since the pressure drop is accompanied by a temperature drop. The exhaust temperature will range from 400 to 750 deg. F.

The return stroke of the piston forces these exhaust gases out through the exhaust line, the gas temperature falling considerably during this stroke. There is, however, the clearance volume, which is filled with burnt gases when the piston reaches the end of its stroke. To remove this volume of inert gas, it is customary to allow the exhaust valve to remain open somewhat after the piston completes this stroke, reverses, and starts on the suction stroke. The admission valve is open ahead of dead center, and the momentum of the column of exhaust gases causes the flow to continue even after the reversal of the piston, creating a partial vacuum into which the fresh air rushes.

Schematic Layout of the Diesel Engine.—Figure 11 embodies the schematic arrangement of the essential mechanism of a Diesel engine. In this particular instance the engine is of the horizontal type operating on the four-stroke-cycle principle. The engine crank is represented with its center at O, and the crank revolves clockwise. To the crankshaft is geared a layshaft A which revolves at half engine speed. On this layshaft are mounted the cams used to actuate the exhaust, admission and fuel-injection valves, which are operated in the sequence outlined in Fig. 8. The fuel pump is driven off the layshaft *A*, the amount of fuel charge being under control of the governor. The fuel is deposited in the fuel valve *C*, out of which it is forced by the air charge at the proper moment. The air blast is supplied by the air compressor *D*, which is driven by a crank on the end of the engine shaft. In this diagram the air line is not supplied with a receiver or air bottle, and the air is delivered directly from the compress or to the fuel valve.

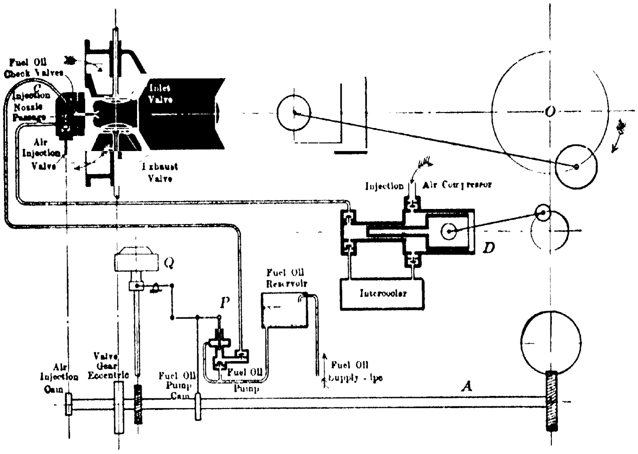

FIG. 11.—Schematic arrangement of Diesel mechanism.

The parts enumerated are the essential parts of the Diesel engine. However the arrangements may differ, it is necessary that the unit include an air compressor, fuel pump, governor, camshaft and injection or fuel valve in addition to those parts generally found on an internal-combustion engine.

Cylinder Events of a Two-stroke-cycle Diesel.—Figure 12 represents the working diagram of a two-stroke cycle. In Fig. 12 *F* the air charge is compressed from *b* to *c*. The fuel is injected when the piston reaches top dead center and the piston forced downward on the working stroke, making the lines *cd* and *de*, Fig 12 *C*; at *e*, Fig. 12 *D*, the piston uncovers passages or ports in the side of the cylinder through which the exhaust gases pass. As the piston moves downward to the point *a*, scavenging valves in the cylinder head open, and a charge of pure air, which has been compressed in a large air pump or compressor to about 4 lb., blows into the cylinder, clearing it of all exhaust gases. At *b*, or slightly before this point, the scavenging valve closes, and as the piston moves upward, it seals the exhaust ports at the point *b*. Continued upward motion compresses the air charge until upper dead center is reached, whereupon the cycle is repeated. In some engines the scavenging valves in the head are replaced by a row of air ports which are uncovered by the piston at the end of the expansion stroke. The air flowing in through these ports clears the cylinder of the exhaust gases.

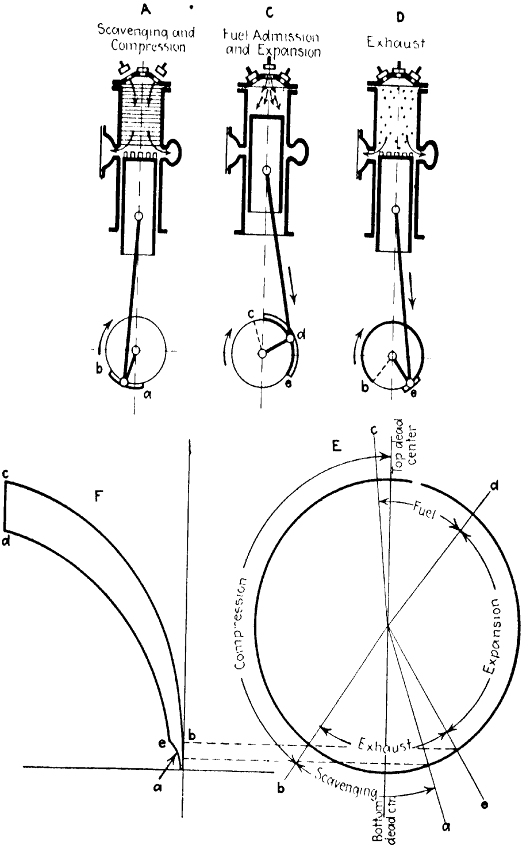

FIG. 12.—Cylinder events of a two-stroke-cycle Diesel.

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CHAPTER IV

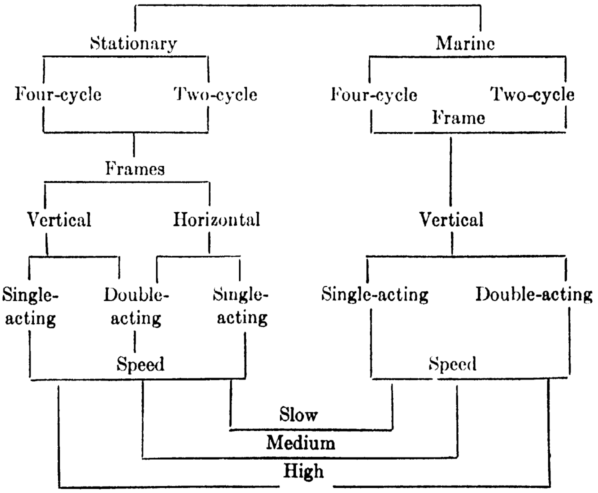
DIESEL ENGINES OF THE UNITED STATES

Classification.—The Diesel engines may be separated into stationary and marine engines according to the service. This classification may be open to objections since a number of engines are used both for stationary and marine service with but little alteration, such as, in the governor arrangement, flywheel, and fuel control. Nevertheless, there is a broad difference in the two classes of engines.

A second natural grouping is the division of the engines into two-stroke-cycle and four-stroke-cycle. If the matter is carried further, we obtain a classification as to design, horizontal and vertical, and also as to speed, the division being medium- and low-speed. The latter class includes speeds up to 150 r.p.m., and the former, 150 to 250 r.p.m., although it is probable that the building of engines of 300 to 600 r.p.m. calls for high-, medium- and low-speed classes.

The complete classification will result in a chart as in Table I.

Table I.—Diesel-engine Classification



Classification by Cycles.—The engines built in the United States, as separated according to the cycle upon which they are operated, two-stroke- or four-stroke-cycle, are shown below.

TABLE II.—BUILDER'S CLASSIFICATION

FOUR-STROKE-CYCLE DIESELS	TWO-STROKE-CYCLE DIESELS
Fulton	Nordberg
Snow	Standard
Allis-Chalmers	Busch-Sulzer
Busch-Sulzer	Bethlehem
McIntosh & Seymour	
De La Vergne	
Winton	
New London (Nelsco)	
Pacific	
Craig	
Cramps	
Worthington	
Dow	
Otto	
New York Shipbuilding	
National Transit	
Lombard	
McEwen	

In Table III are shown the engines of the several Diesel manufacturers separated as to type and service.

TABLE III.—SERVICE CLASSIFICATION

STATIONARY	MARINE
American Diesel	New London
Busch-Sulzer	McIntosh & Seymour
Snow	Busch-Sulzer
McIntosh & Seymour	Pacific
Allis-Chalmers	Craig
Fulton	Cramps
McEwen	Winton
National Transit	Nordberg
Standard	Bethlehem
De La Vergne	Worthington
Nordberg	New York Shipbuilding
Dow	Dow
Winton	
New London	
Otto	
Lombard	
Bethlehem	

TABLE IV.—CLASSIFICATION BY FRAME DESIGN

VERTICAL	HORIZONTAL
Nordberg	Snow
McIntosh & Seymour	Allis-Chalmers
Fulton	McEwen
Worthington	De La Vergne
Busch-Sulzer	Otto
Standard	National Transit
Pacific	Standard
New London	
Snow	
Craig	
Cramps	
Winton	
Dow	
New York Shipbuilding	
American Diesel	

Diesel Engines Built in the United States.—Before discussing to any great detail the several engines built in this country, the author has taken the liberty to give short thumb sketches of each engine. This is with the idea of allowing the reader, if unacquainted with the industry, to gain a mental picture of the general appearance of the product of each manufacturer.

Stationary Diesel Engines.—Since stationary Diesels are in the majority, this type will be discussed first.

American Diesel Engine Company.—This company was the pioneer Diesel engine company in the United States. At the present time there are many installations of the American Diesels which have been in operation for fifteen to eighteen years, though they have in many instances been replaced by units of modern design and are kept for standby service. For this reason the engine is mainly of historical interest although there is more demand for experts to adjust and repair these engines than any other. This, of course, is because the engines have been in service for such lengths of time that extensive overhauls and rebuildings are necessary.

Figure 13 is the cross-section of the American engine. Even to the experienced it is evident that the builders largely followed accepted gas-engine designs in the general construction of this unit. The frame was of the box type and was reinforced by tension rods to absorb the working stress. The cylinders, of which there were three per engine, were cast integral with the jacket walls. The valves were located at the side of the cylinders quite like the gas engine practice of 20 years ago. The box frame was entirely enclosed, though provided with side doors, and oil and water splash lubrication was used. The camshaft was carried inside of the frame in bushed bearings bolted to the interior of the frame; the gear reduction was 2 to 1, with an idler pinion between the camshaft and the crankshaft gears. The

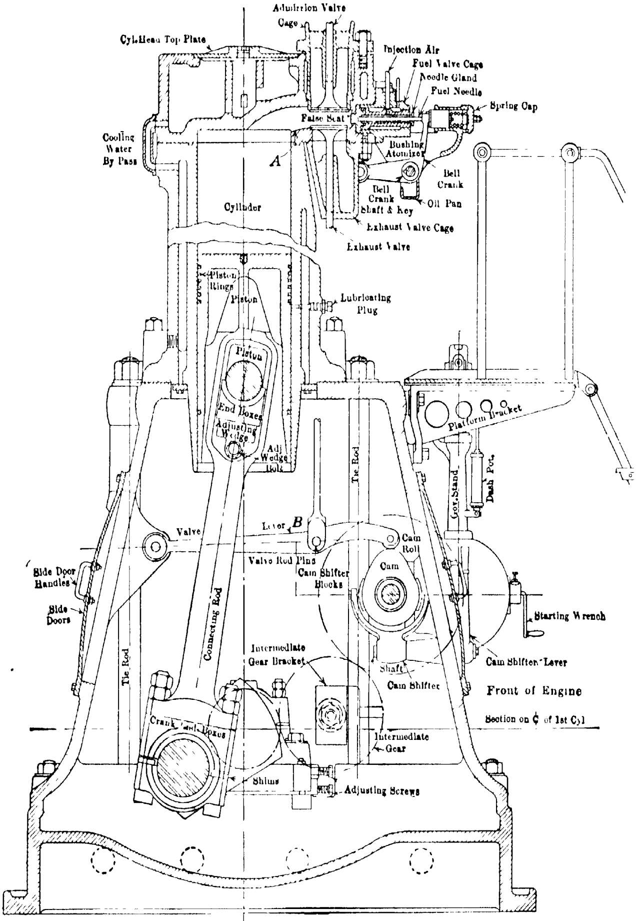


FIG. 13.—Cross-section of American Diesel engine.

various parts will be discussed in succeeding chapters as will also the parts of other engines.

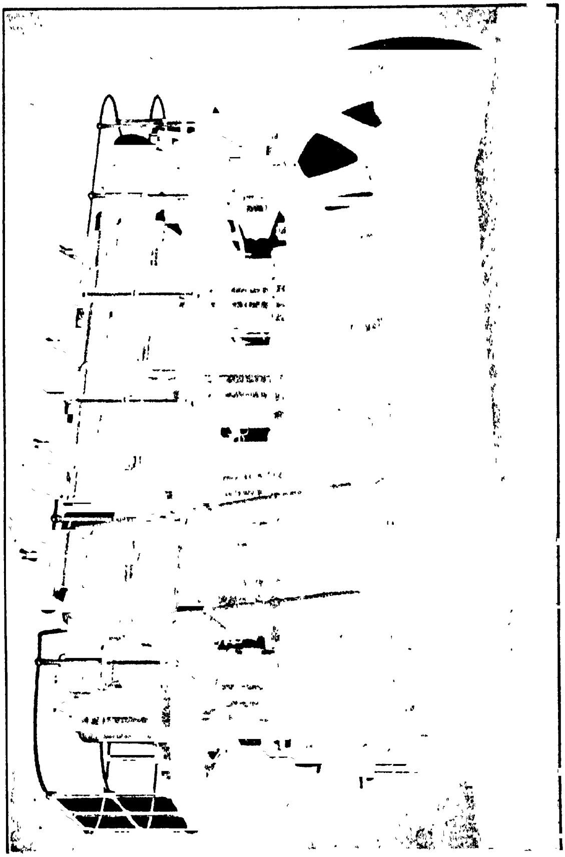


FIG. 14.—520 hp. Busch-Sulzer Type *B* Diesel.

The engine, when first introduced, was of a type unknown to the American engineer and, like all new machines, suffered at the hands of ignorant and unskilled laborers. Probably no prime mover ever experienced the manhandling accorded this oil engine. Scores of cylinder-head stud bolts were twisted off under the efforts of brawny laborers using a heavy wrench and sledge. As faulty as the engine was, it is highly probable that it would have been more successful under the attention of skilled operators. It should not be presumed that all these engines have been scrapped. To the contrary, the majority are in daily operation for a few hours a day, while scores are on light work. As an example, one engine has been in constant service 20 hours per day since 1905.

FIG. 15.—750 hp. Busch-Sulzer two-stroke-cycle Diesel.

These original engines, however, served the useful purpose of giving many Diesel engineers experience and caused American engineers to think in terms of the Diesel engine. Later-day manufacturers should, for this reason, have a grateful feeling toward this pioneer.

Busch-Sulzer Bros. Diesel Engine.—This company represents a reorganization of the American Diesel Engine Co. In the design of the Type B four-stroke-cycle stationary engine the experience of their Swiss associates, Sulzer Bros., has been drawn upon. The engine, a view of which appears in Fig. 14, has four working cylinders, while the air compressor is mounted on the engine frame and has much the appearance of a fifth cylinder. The frame is of two-piece construction, the base carrying the shaft bearings The upper portion of the frame rests on the base and is tied to it both by base stud-bolts and tension bolts or tie rods.

The camshaft is mounted close to the top of the cylinder and is driven by a vertical drive shaft, which also carries the governor. The camshaft is entirely enclosed, the cams working in an oil bath. Doors provide access to the various parts. The valve mechanism has incorporated with it a servomotor, which controls the injection point and the air-injection pressure. The cam housing also includes the support for the fuel pumps which are driven from off the vertical cam drive shaft. This company builds its larger engines on the two-stroke-cycle principle, as shown in Fig. 15.

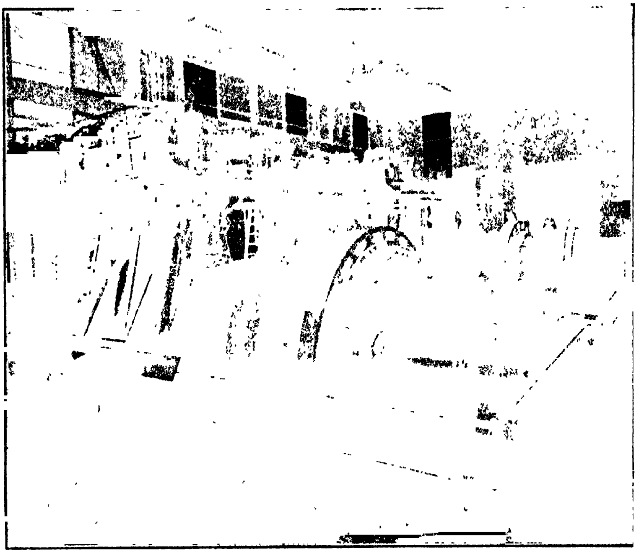


Fig. 16.—500 hp. McIntosh & Seymour Diesels, Texas Light and Power Co., Paris, Texas.

McIntosh and Seymour A-frame Diesel.—The McIntosh and Seymour Corp. secured the American license of the Aktiebolaget Diesels Motorer (Swedish Diesel Engine Co.) of Sweden. Following the practice of the Swedish manufacturers the American firm first made use of the A-frame design.

As outlined in Fig. 16 the engine frame or base is a low box section casting some 12 in. in height. This frame rests directly on the foundation. To the base are bolted the A-frame castings, which are of cylindrical form at the top to receive the cylinder liner, in this way acting as the cooling jacket outer walls. In the 500 hp. unit there are four cylinders and A-frames, while the air compressor is mounted in line with the power cylinders on a similar A-frame but of much lower height. The 1,000-hp. unit has six cylinders and frames.

The camshaft is carried at a level with the cylinder heads and is driven by a vertical shaft and bevel gears; the shaft carries the governor while the fuel pump is supported on one of the cylinder castings.

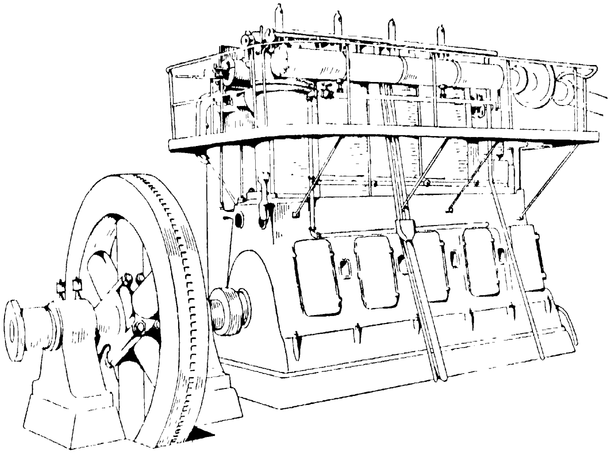


Fig. 17.—McIntosh & Seymour Type *B* box-frame Diesel.

The McIntosh & Seymour Corp. has abandoned the A-frame engine for stationary work and are centering their manufacturing facilities on the box-frame engine. Beyond the frame itself, the engine has been modified in no great way, the details of valves, governors, etc., remaining as before. Figure 17 is a view of the box-frame engine. These engines range in size up to 1,000 b.hp., which unit is of six-cylinder construction.

McIntosh & Seymour Large Diesels.—This company builds a crosshead type vertical Diesel in sizes ranging from 750 to 2,200 hp. This engine, illustrated in Fig. 18, has the camshaft running alongside the engine at a level with the cylinder bases, the camshaft being driven by a train of spur gears from the engine shaft.

The valves are actuated by rocker arms and reachrods. The reachrods carry rollers which contact with cams on the camshaft. The valves are in no way different from those of the type A engine

The cylinders are supported by box girders consisting of a front and rear column cast together with a top cross member. One of these frames supports the adjacent sides of two cylinders.

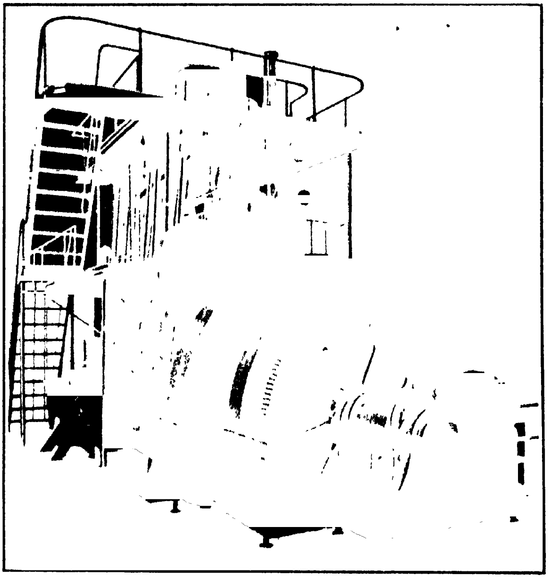


FIG. 18.—750 hp. McIntosh & Seymour crosshead-type Diesel.

Worthington Pump and Machinery Co. Diesel Engine.—This corporation manufacturers, at the Snow Works, a Diesel-type engine under the name of "The Snow Oil Engine." This engine, which is of horizontal construction, is manufactured in units from 65 to 800 hp. It does not follow standard American Diesel practice for medium-powered engines in that a crosshead is used in preference to a trunk piston. This necessitates the lengthening of the frame.

The main frame, Fig. 19, is of box section with both longitudinal and transverse ribbing. The single and twin-cylinder units have a one-piece frame, while the frame of the three-cylinder engine is of two-part construction to facilitate shipping and erection. As seen by Fig. 19, the frame is extended to form the cylinder cooling jacket; the liner is separate and is a press fit into the jacket housing or casting. The air charge for the engine cylinders enters the frame at one side of the crankcase and flows along the frame before passing through the admission valve.

FIG. 19.—Snow horizontal Diesel.

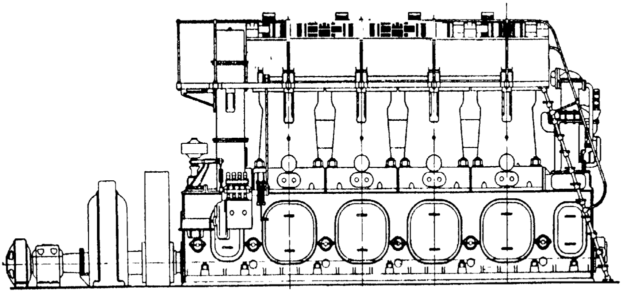


FIG. 19A.—Snow vertical engine.

The air compressor is mounted on the engine frame and is driven by a crank keyed to the engine shaft. The valves are controlled by a camshaft placed transversely in front of the cylinder head; this shaft is driven through bevel gears by a layshaft at one side of the engine. This latter shaft also drives the governor and fuel pumps.

Snow Vertical Diesel.—The Worthington Pump and Machinery Corp. has designed a line of vertical trunk type engines for stationary service at the Snow Works. The 750 hp. unit is shown in Fig. 19*A*. The frame is of the one-piece box type and is mounted on a cast-iron subbase. The cylinders rest upon the top surface of the bed. The camshaft lies along the cylinder tops, the valves being driven by their respective cams by short rockers.

Allis-Chalmers Diesel Engine.—The frame is of the box type, Fig. 20, and is so designed as to allow the engine shaft to rest deep in the bearings. This gives the engine a low center of gravity, making it rigid and fairly free from vibration while in operation.

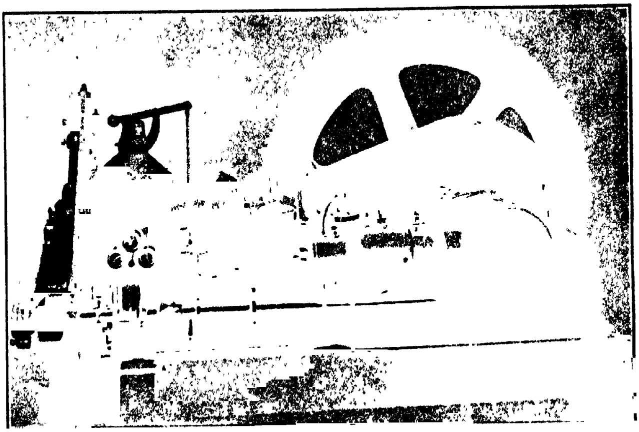


Fig. 20.—Allis-Chalmers Diesel.

As with practically all horizontal Diesels the frame forms the cylinder jacket, while the liner is held in place by a flange at the head end. The valve mechanism is driven off the longitudinal layshaft, which also handles the governor and fuel pump. In later engines a camshaft is run across the front of the cylinder heads and driven by the layshaft. The Allis-Chalmers engines are manufactured in single-, twin-, triple- and quadruple-cylinder units. The single and twin engines are both built with a one-piece frame. The triple-cylinder unit consists of a single and a twin engine with the flywheel between the two engines, the two-piece shaft being flanged and bolted to the wheel hub. The quadruple-cylinder engine is obtained by using two twins with the flywheel between the two frames.

National Transit Pump and Machine Co. Diesel Engine.—The engine manufactured by this company was of horizontal design, as outlined in Fig. 21, which brings out the massive lines, particularly those of the bearing housing footings. The engine was built in one-and two-cylinder units. As usual with the horizontal design, a layshaft is employed to control the valve mechanism, as well as to drive the pump and governor. Not a great number were built (approximately 15), and the manufacturers centered their activities upon the construction of oil-pipe-line-pumps which have always been their chief product.

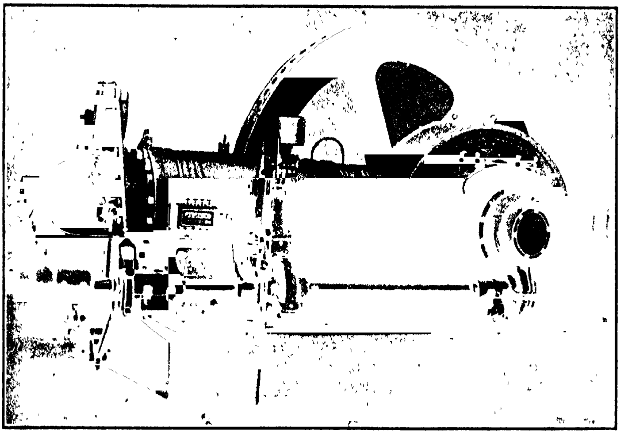


FIG 21.—National Transit engine.

McEwen Bros. Diesel Engine.—This company was one of the early firms to embark upon Diesel-engine manufacture. The engine closely resembles other American units, being of the horizontal design, Fig. 22; the cylinder liner is pressed into the jacket, which is a part of the frame. The piston is of the standard trunk design. The valves are vertical, and both admission and exhaust valves are provided with cages. The valves are controlled by separate eccentrics while the injection valve is actuated by a cam. The air compressor, which is two-stage, is driven by a

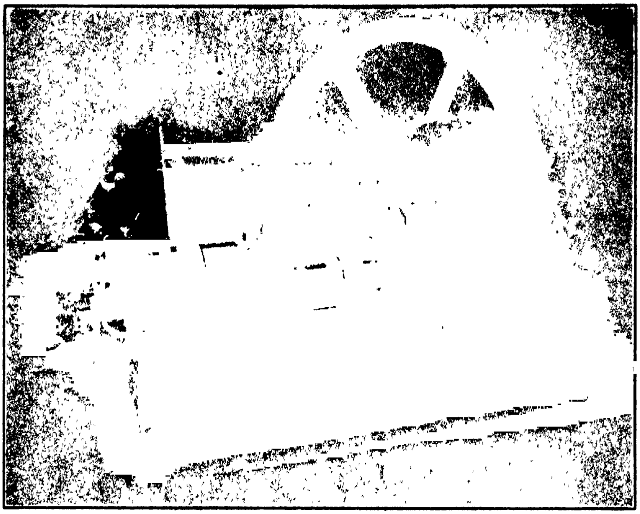


FIG. 22.—McEwen Bros. Diesel.

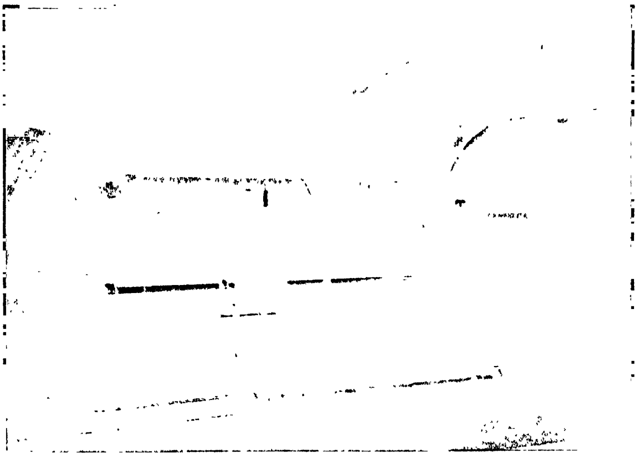


FIG. 23.—De La Vergne F. D. Diesel.

crank extension of the engine shaft. No air storage is employed, a small bottle being placed in the air discharge line to absorb the air pulsations. A dozen or more of these engines were built, and the company then engaged in the building of semi-Diesel engines.

De La Vergne Machine Co. Diesel Engine.—The De La Vergne Machine Co. has been manufacturing oil engines of the low-pressure and the semi-Diesel types for a number of years. In 1918 they brought out their Type FD engine, which operates on the true Diesel cycle. The engine, which appears in Fig. 23, is of horizontal design, and embodies the use of a very rugged frame. The camshaft is borne in bearings in front of the cylinder head and is driven by the longitudinal layshaft through bevel gears. The valves lie horizontally and are actuated by short cam levers. The fuel pump and governor, which are driven by the layshaft, are similar to those used on the De La Vergne FH engine and are the result of a number of years of experience.

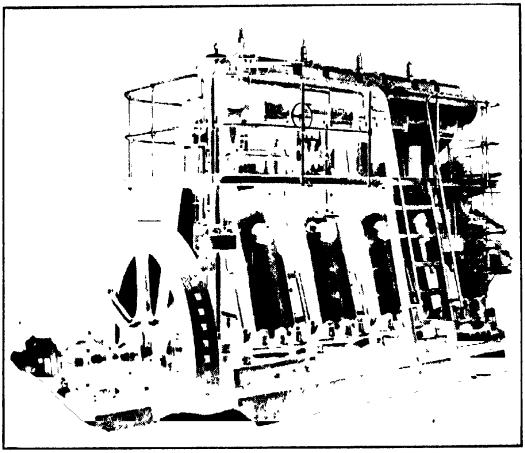


FIG. 24.—Fulton-Diesel.

The development of the solid-injection engine induced this firm to abandon the air-injection engine, and it now builds only semi-Diesel and solid-injection oil engines. The latter engine is discussed in Chapter XX.

Fulton Iron Works Co. Diesel Engine.—The Fulton Works Co. has an Italian Tosi license to build stationary Diesels. The engine illustrated in Fig. 24 is of the A-frame type with the cylinder liners pressed into the frame housing. The camshaft runs along the side of the cylinder heads and is driven by a vertical drive-shaft to which is also geared the governor. The engine has some features quite different from other American Diesels. The governor controls the stroke of the fuel pump plunger rather than the period of lift of the suction valve. The air starter embodies no mechanically operated air valve as is standard with other engines. A distributor mounted on the

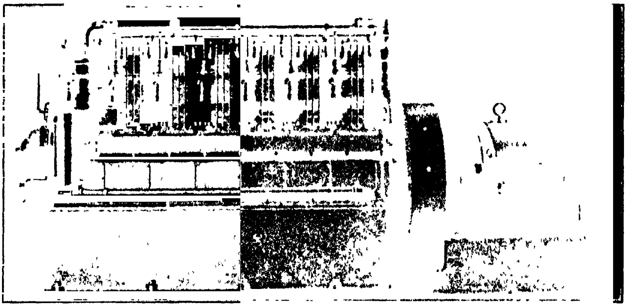


FIG. 25.—Winton engine-generator set. (Six cylinders, 7½ × 11 in., 450 r.p.m.)

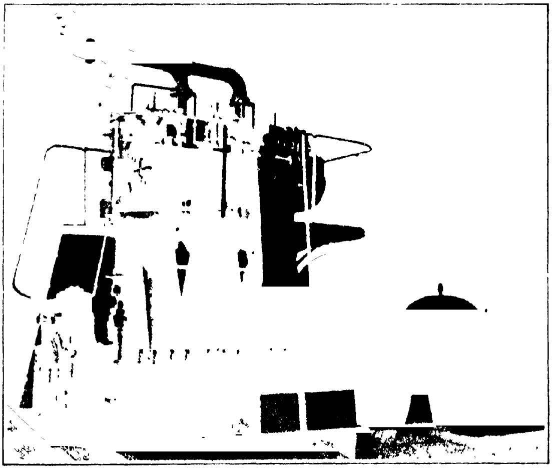


FIG. 26.—Dow-Willans Diesel.

camshaft allows the starting air to pass to the proper cylinder. The Fulton engine is built in sizes from 120 to 750 hp.

Winton Engine Co. Diesel Engine.—The Winton Engine Co., the founder of which has been prominently identified with the automobile industry for many years, in 1918, brought out a vertical high-speed Diesel engine, in the design of which is incorporated a number of new ideas. Figure 25 shows an exterior view of the engine direct connected to a generator, which arrangement is also followed in the marine engine with the propeller motor-driven.

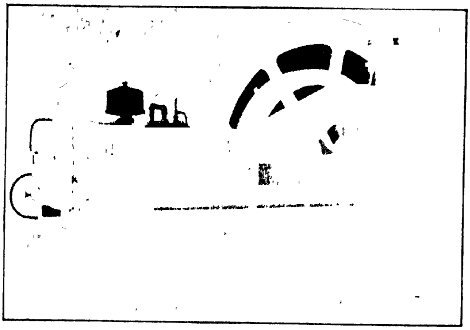


Fig. 27.—Otto oil engine.

Dow Pump & Machinery Co. Diesel Engine.—The American rights of the Willans Diesel, an English engine, were acquired by the Dow Pump Machinery Co. The engine is of the vertical A-frame type and was used for both stationary and marine service. No unusual features distinguished the unit from others, with the exception of the use of the Reavall compressor which as a separate unit is connected to the crankshaft extension.

Otto Engine Co. Diesel Engine.—The Otto Engine Co. has long been engaged in the manufacture of gas and gasolene engines and has built a few Diesels. These units follow the general design of the horizontal Deutz Diesels, although much of the fuel valve and pump is original.

Atlas-Lyons Diesel Engine.—Although only five Atlas-Lyons Diesels were built, the engine is shown in Fig. 27*A* as a matter of historical interest. The main bearings were provided with wedge adjustment; piston-water cooling was used and the fuel spray entered the combustion chamber at an angle. As far as is known, only one of these units is in operation at present. The engine represents an attempt to combine a number of European designs into a distinctly American engine.

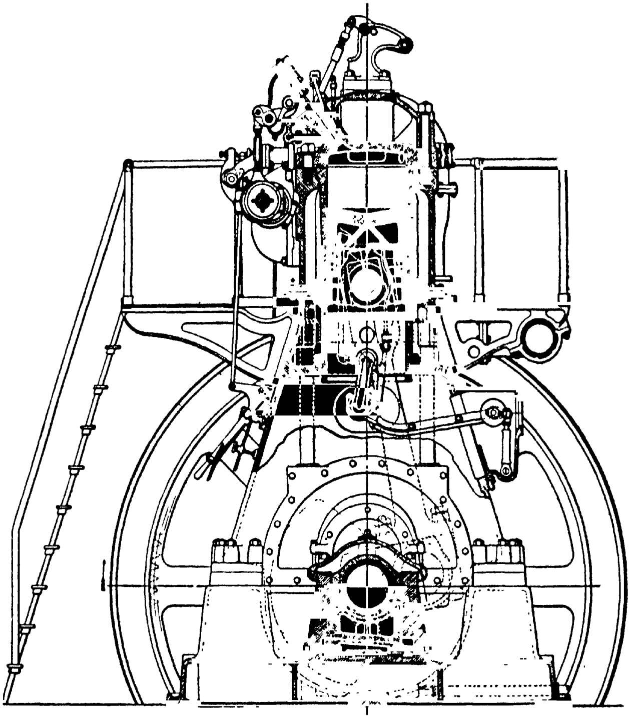


FIG. 27A.—Lyons-Atlas engine. Original design.

Lombard Diesel.—Figure 28 illustrates the vertical four-stroke-cycle Diesel manufactured by the Lombard Governor Co. This engine is somewhat a departure from current American design. The base is of the box type, supporting the cylinders. Each cylinder is cast in one piece with the cylinder head, in this way avoiding the use of a gasket as is usual at the juncture. A still more important advantage claimed by builders is that the construction eliminates the thickening of the metal at the top of the cylinder. It is generally recognized that this thick section has caused trouble by reason of excessive heating.

The layshaft runs along the top of the engine and is driven by the vertical governor shaft. The valves are set into the cylinder head at an angle, while the fuel valve enters vertically on a line with the cylinder bore. The oil is sprayed horizontally into the narrow space between the piston and cylinder head.

At the present time the builder is making radical changes in the design.

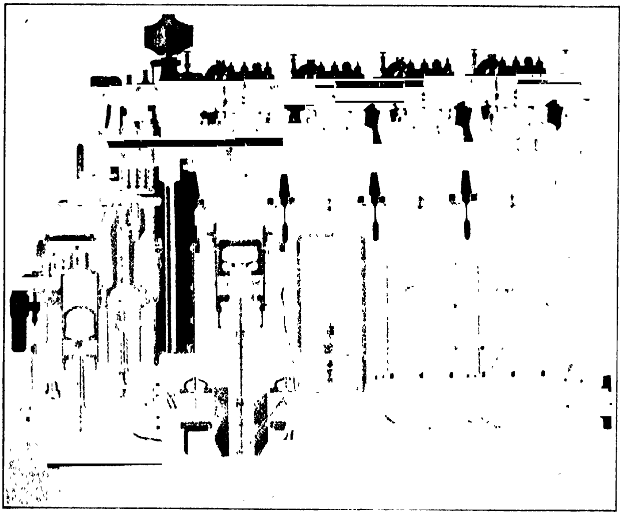


FIG. 28.—Lombard Diesel.

New London Stationary Diesel.—The New London Ship & Engine Co. entered the stationary field recently although they have manufactured marine Diesel for years. The 600 hp. New London four-stroke-cycle unit is shown in Fig. 29. The engine is provided with a box frame which rests on a girder subbase. The frame is extended to form the jacket for the cylinder liners which is pressed into the jacket, being sealed by a rubber gasket at the lower end and by a flange at the upper face.

The combustion space, unlike that in many engines, does not consist of a flat cylindrical volume between a flat piston head and a flat cylinder head. To overcome the objectionable feature of this type of combustion chamber, which presents a large cooling area per unit of volume, the New London engine has the combustion space located in the cylinder head. This gives a maximum combustion volume per unit of surface area. The cooling effect is reduced, and since the air charge is concentrated about the fuel valve, the mixing of the fuel and the air is improved.

To operate the several valves, a layshaft is placed along the engine at a level with the top of the cylinders. This layshaft carries the required valve cams and is driven by a train of gears from the crankshaft.

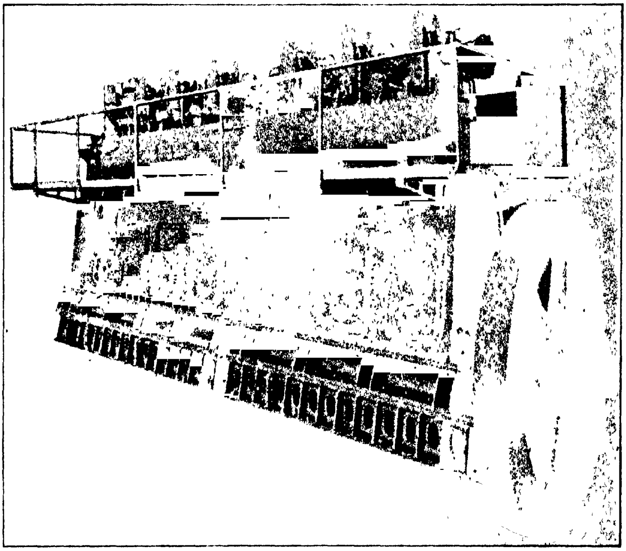


Fig. 29.—New London 600 hp. unit.

Two-stroke-cycle Stationary Diesels.—The American manufacturers, with but few exceptions, have adopted the four-stroke-cycle engine. In this they were undoubtedly influenced by the greater freedom from operating difficulties which this type possesses over the two-stroke-cycle as built several years ago. At present the majority of the large stationary engines are two-stroke-cycle and are as reliable as a four-stroke-cycle unit.

The two-stroke-cycle, chiefly on account of the lighter weight per horsepower, has been in favor in marine work. It has not won complete possession of this field, and at present the swing is strongly toward the four-cycle. For the small-powered boat under 200 hp. rating and for the large motorships calling for engines of 2,000 hp. or greater, the two-stroke-cycle engine is better adapted than is the four-stroke-cycle. This applies especially to submarine boats, even though the latter engine is more popular in this field at the present time.

Nordberg Two-stroke-cycle Diesel Engine.—The Nordberg Manufacturing Co. obtained the rights to build the Carels (Belgium) two-stroke-cycle Diesels. The Nordberg Co. displayed a great deal of boldness in building 1,200 hp. Diesels even though the other American firms were loath to offer engines larger than 500 hp. As a demand existed for larger units in the mining field, Diesels of 2,000-b.hp. capacity each were built by the Nordberg Co. and put into service a considerable time before other manufacturers discerned the drift toward large units. Up to the present time eight 2,000 hp. units have been placed in active service in the mining industry. While the total horsepower output of Nordberg Diesels does not equal that of three other firms, their engine capacity in units above 1,000 hp. places this company in the front. The large Nordberg engines are somewhat unusual having four cylinders rather than six as is customary with other large Diesels of 2,000 hp. This engine appears in Fig. 30.

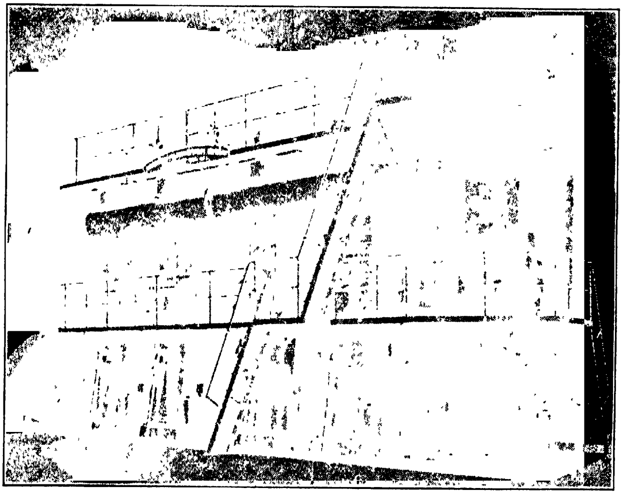


Fig. 30.—Nordberg Type *F.D.* 2,000 hp. engine.

Nordberg Type V.E. Diesel.—A two-stroke-cycle Diesel of small horsepower, 110 hp. per cylinder has been built and developed by the Nordberg Manufacturing Co. The engine is of slow speed and is nicely adapted for standard sizes of alternating current and direct current generators. It is a two-stroke-cycle port-scavenging crosshead type engine with open frames, so that the reciprocating parts are readily accessible.

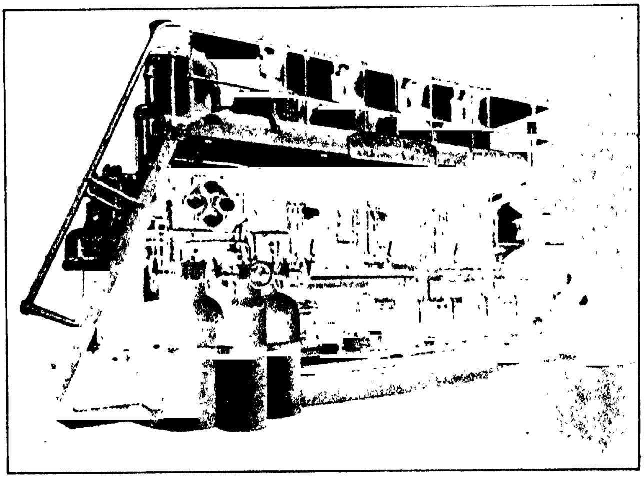


FIG. 31.—Nordberg Type *V E.* two-stroke-cycle Diesel.

The engine has a direct-driven three-stage air compressor and a double-acting air pump to furnish the necessary starting, blast and scavenging air. Compressor and scavenging pumps are driven by separate cranks from the crankshaft. Water cooling is provided for the cylinders, cylinder heads, crosshead guides, pistons and exhaust pipes.

The cylinder and frame are cast in one piece and a liner is fitted in to form the cylinder proper. The liner is easily removable and heavy enough for reboring. The cylinder head is made of special cast iron and is designed to make a symmetrical casting of as near uniform thickness as possible. The only opening in the head is the one for the fuel atomizing and starting valves. The piston is of the two-piece type, the head is water cooled and contains the piston rings, while the trunk which is attached to the lower end of the piston head is fitted comparatively loose to

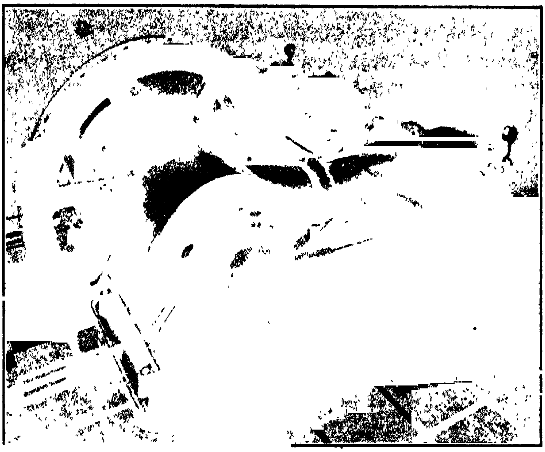


FIG. 32.—Standard two-stroke-cycle engine.

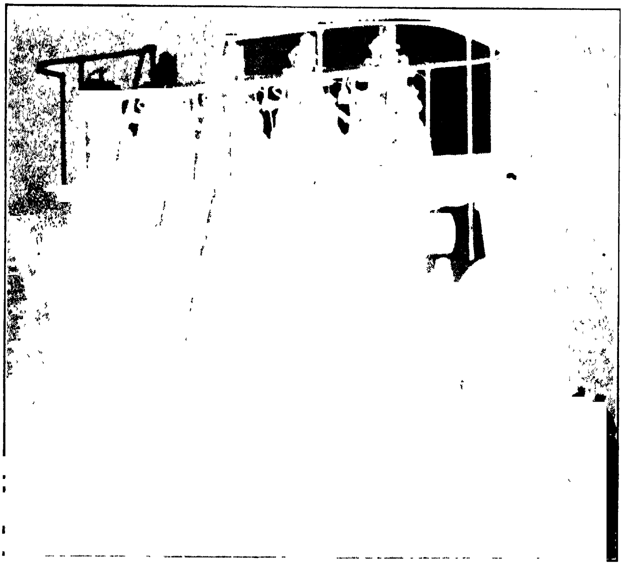


FIG. 33.—Standard vertical two-stroke-cycle type.

eliminate any danger of seizing; this construction being possible only with the engine of crosshead type.

Standard Two-stroke-cycle Diesel.—The Hadfield-Penfield Co manufactures the Standard two-stroke-cycle Diesel in both

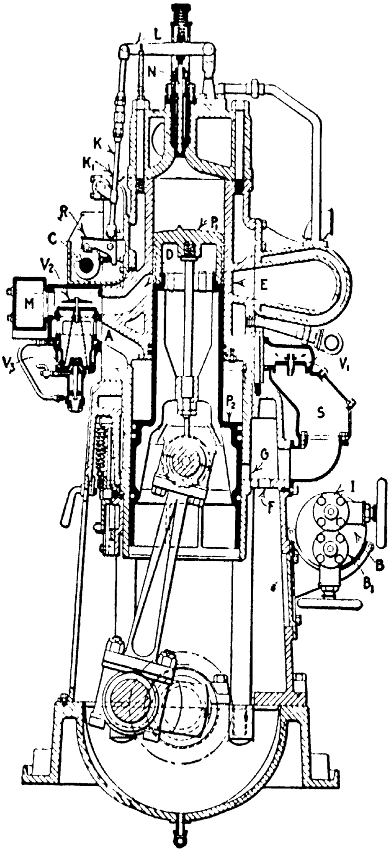


FIG. 34.—Southwark-Harris two-stroke-cycle Diesel.

vertical and horizontal types. The Standard was the first two-stroke-cycle Diesel built in the United States, and considerable credit is due to the manufacturer and to the designer that they persisted in holding to the two-stroke-cycle principle in the face of the general adoption of the four-cycle idea.

Originally the Standard was built only in the horizontal type, Fig. 32, but in 1921 the vertical design, Fig. 33, was brought out, and this is the type now offered for sale in general.

The Southwark-Harris Diesel Engine.—The Southwark Foundry & Machine Co. manufactured a limited number of two-stroke-cycle engines that were applicable both for stationary and marine work. This engine, a cross-section of which is shown in Fig. 34 is of vertical design and has cylinders ranging in number from four to eight, dependent on the horsepower rating and is here described briefly since it is no longer made.

Busch-Sulzer Bros. Two-cycle Diesel.—As mentioned heretofore, this company manufactures all of its engines above 500 hp. to operate on the two-stroke-cycle. One of the engines is shown in Fig. 15.

MARINE DIESEL ENGINES

General.—The Diesel engine is especially attractive as a marine prime-mover since its fuel consumption by weight is approximately one-fourth that of a steam plant. This advantage is not solely one of lower fuel costs but includes the item of lessened bunker space. The cargo room of the motorship is much greater than that of a steamer of the same tonnage.

There has been a large number of motorships built in Europe. In the United States the development has not been as great as one would expect. This is attributed to the lack of progressiveness on the part of shipowners and to the large tonnage of steamers built during the war. With this enormous tonnage of vessels lying in harbor idle the marine concerns have been unwilling to invest additional money in motorships even though the operation costs would be much reduced. Unfortunately these steamers cannot be changed to Diesel drive without expensive alteration to the hulls.

It has been a matter of comment and of much criticism that the United States Shipping Board has been so disinclined to consider the Diesel for the Emergency Fleet Vessels. As an example of the dilatory tactics of the Board, the case of the *William Penn* may be cited. Burmeister and Wain Diesels were imported from Denmark and lay in storage practically three years before being installed in the vessel. Little encouragement has been given to American engine builders and these concerns are to be commended for their progressiveness in bringing out marine engines in spite of such official lassitude.

In the last two years a number of Diesels from 300 hp. to 3,000 hp. have been installed in motorships. The shipbuilding has been chiefly confined to tankers operating between Mexico and other ports. Since Mexican bunker oil is very cheap, it is the claim of some shipowners and builders that the steamer is the most economical for such service, for the average Diesel requires a higher grade oil than bunker oil. Engine manufacturers and shipowners anticipate an increased use of Diesels in vessels to be built in the future.

Nobel Bros. Marine Diesel. The honor of building the first commercial line of marine Diesels belongs to this Russian firm. Since the demand was for Diesels to engine the river and ferry boats, the units Nobel Bros. produced were all of small powers—ranging from 50 to 200 hp. The same firm was the first to adapt the Diesel engine to submarine boats. The Russian government leaned to the small-dimensioned subboats, consequently the development of the Nobel Bros. submarine Diesel has been along the smaller units below 200 hp., although they produced a 900 hp. submarine Diesel prior to 1914. Like all new apparatus the Nobel Diesel has undergone various changes in design.

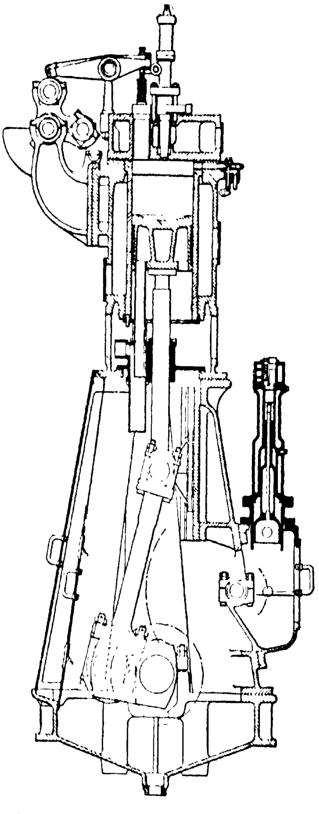


FIG. 35.—Cross-section of Werkspoor Diesel on the motor-ship "Vulcanus."

Werkspoor Marine Diesel.—The first ocean-going motorship of any size equipped with a Diesel engine was the *Vulcanus*. This boat was a 1,000-ton tank ship fitted with a 450-hp. Werkspoor engine having cylinders 15.7 in. diameter by 31.5 in. stroke, operating at 180 r.p.m. This engine is shown in Fig. 35. Although the design necessitated an increased height, the crosshead type piston was used. To eliminate the piston-heating risk, water-cooling through telescopic tubes was used. The frame was of box design and was early abandoned for the present Werkspoor frame which consists of steel columns. This later construction is illustrated in Fig. 36. This engine is being manufactured in the United States by the Pacific Skandia Co., and the New York Shipbuilding and Engine Co. also have a license and have built two 1,750 hp. engines to be installed in a tramp boat.

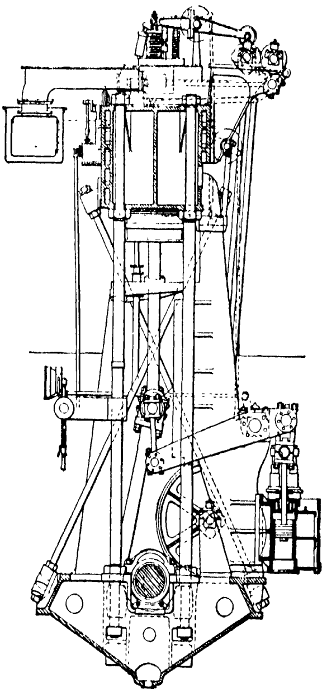


FIG. 36.—Werkspoor modern open-frame marine.

Pacific Diesel Co.'s Marine Diesel.—The Werkspoor Diesel as manufactured by the Pacific Diesel Engine Co. departs somewhat from its European prototype. This engine is shown in Fig. 37.

Nelseco Marine Diesel. The New London Ship & Engine Co. manufactures both two- and four-stroke-cycle marine Diesels.' Figure 38 is a cross-section of the latter type engine. This engine is built both reversing non-reversing.

The largest part of the New London output has been installed in the United States Navy, it being the first Diesel used for submarine purposes. The cylinder and cylinder heads are cast together. This eliminates a gasketed joint and the heating effect which often occurs at the top of the cylinder where the metal is thickened to hold the cylinder-head bolts. However, piston removal is made difficult by this construction. Figure 38 illustrates the non-reversing engine, which differs from the small reversible Diesel in no way other than in the provision of a reversing control described in the chapter on Valves.

McIntosh & Seymour Marine Diesel.—The first marine Diesels manufactured by the McIntosh & Seymour Corp. were built in 1917 and were of the trunk-piston box-frame type, illustrated in Figs. 39 and 40. These engines, 300 and 690 hp. capacities, were equipped with direct reversing gear.

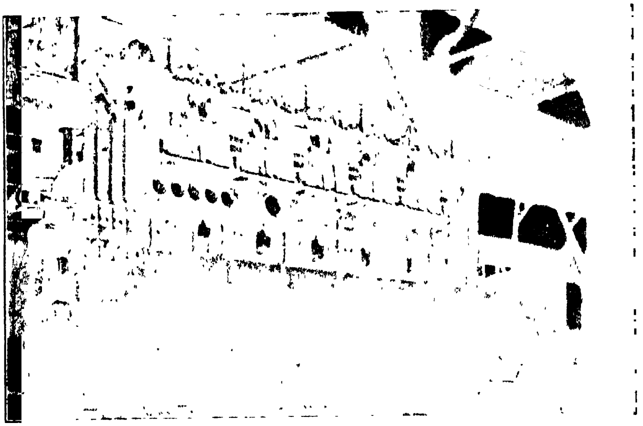


FIG. 37.—Pacific Werkspoor marine engine.

Later, larger units of 900, 1,250 and 1,750 hp. were built and other than the reversing cams and controls, is the same design as the engine in Fig. 18. For these engines the crosshead design is used; the framing consists of cast-iron box girder A-frames upon which supports the cylinders are mounted. The A-frames, in turn, are bolted to a cast-iron base which forms the crank oil pan. Light steel covers fit between the frames, enclosing the crankshaft and other parts.

The majority of the Diesel engines used in the merchant marine in the United States have been built by the McIntosh & Seymour Corp.

Busch-Sulzer Marine Diesels.—The marine engine designed by the Busch-Sulzer Bros. Co. is of the two-stroke-cycle type and is in the main the same as the Sulzer Bros. Marine Diesel as

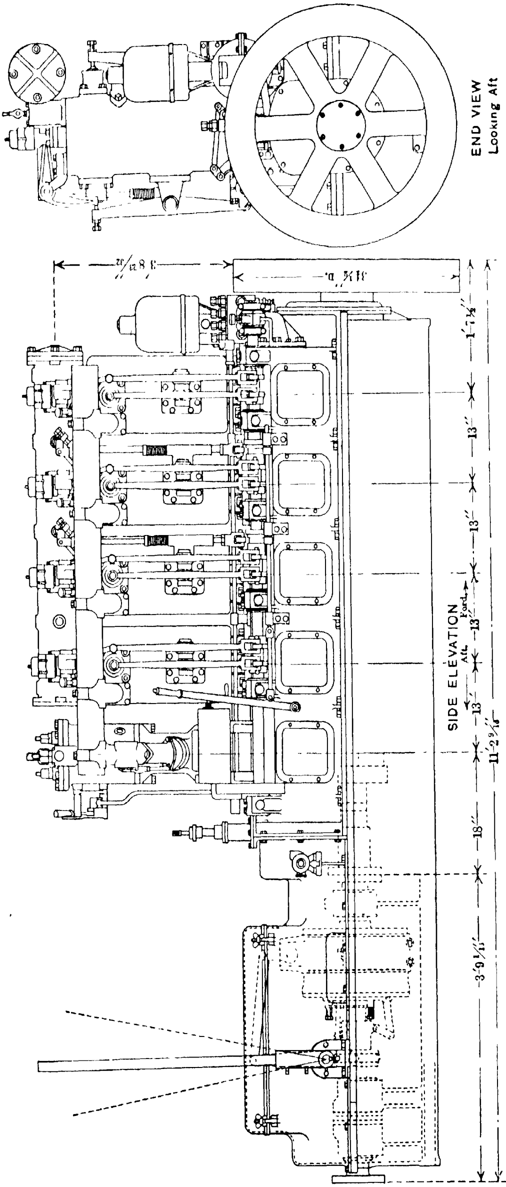


FIG. 38.—New London Ship & Eng. Co. marine Diesel.

manufactured in Switzerland. This engine is shown in Fig. 40 *A*.

During the war this company built many submarine Diesels for the United States Navy. These engines are of the high-speed four-stroke-cycle type and represent the most modern of design.

Cramp Engine & Shipbuilding Co.—The Cramp Co. manufactured a marine Diesel under the Burmiester and Waine patents. One of these units, a 1,750-b.hp. engine, is shown in Fig. 41. The cylinders are mounted on cast-iron box columns, the front and rear columns, between each pair of cylinders being cast together. The crankshaft is built up, the pin being shrunk into the webs and the webs shrunk and keyed to the shaft portions.

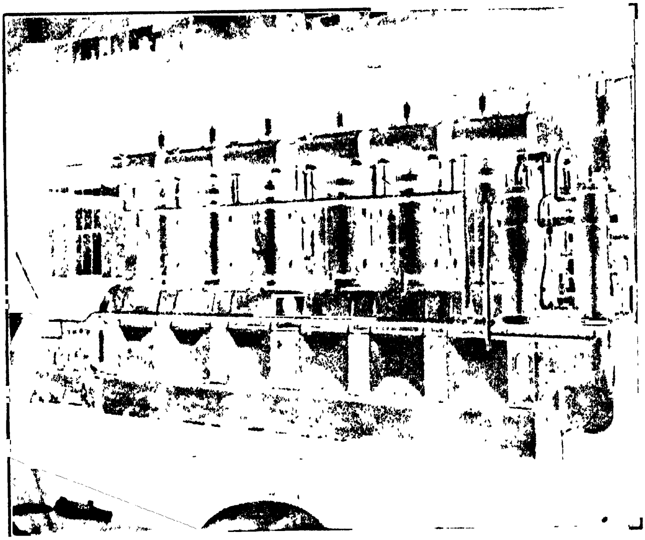


FIG. 39.—McIntosh & Seymour 600 hd trunk-piston pesign.

Winton Engine Co. Marine Diesel.—The marine Diesel built by the Winton Co. is the same as its stationary engine, illustrated in Fig. 25.

Worthington Pump & Machinery Corp.—The Worthington Pump & Machinery Corp. has built at the Snow Works a 1,750-b.hp. six-cylinder marine Diesel shown in Fig.41 *A*. This engine embodies many tried features, such as, mushroom-headed fuel

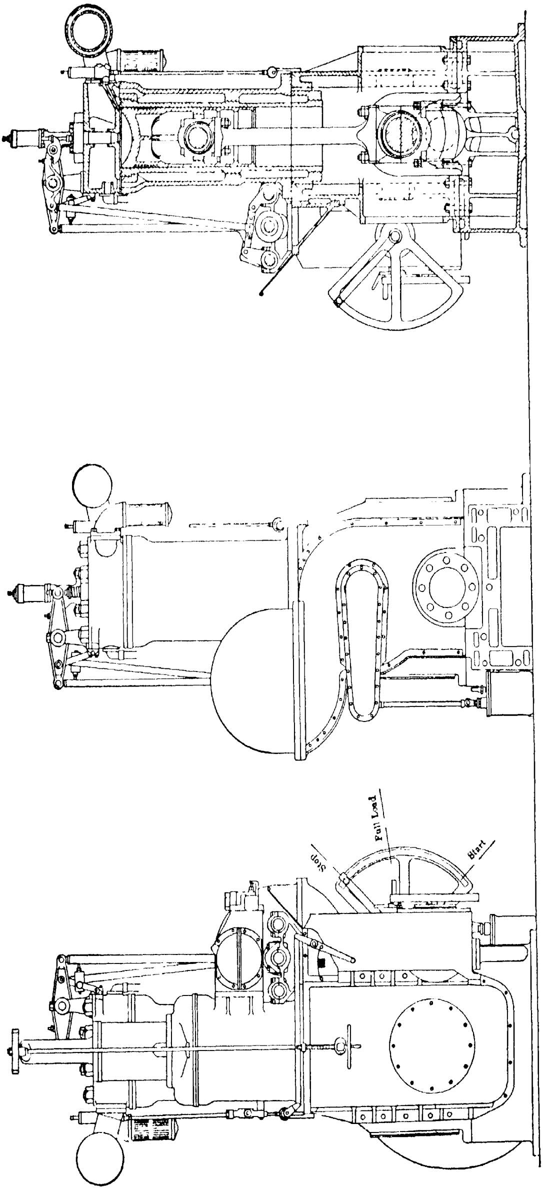


FIG. 40.—McIntosh & Seymour 640 hp. trunk-piston marine Diesel.

valve, reversing control, etc. Probably the most elaborate tests ever carried on in the United States have been run with this unit. Among others a number of tests were made with the engine using auxiliary exhaust ports in connection with the ordinary exhaust valves. While none of these engines have so far been installed on board ship, the engine undoubtedly has a great future.

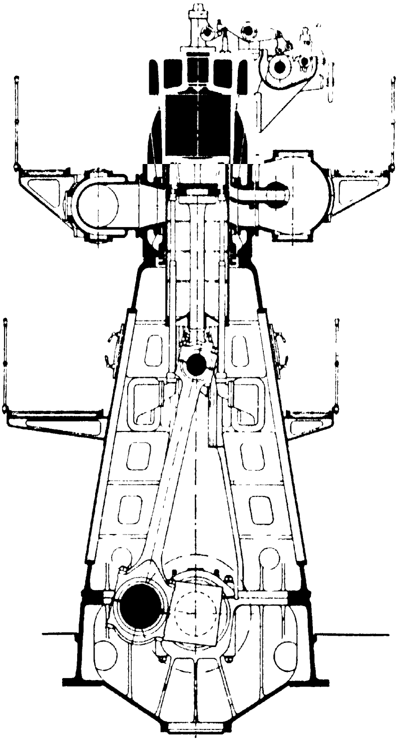


FIG. 40A.—Cross section of Busch-Sulzer marine design.

Nordberg Mfg. Co. Marine Diesel.—The Carels Diesel is offered for marine work although so far none have been installed.

Bethlehem Steel Co. Marine Diesel.—The Bethlehem Steel Co. has built one 2,700-b.hp. Diesel of the two-stroke-cycle design. This is installed on the Cubore, an ore boat plying between Cuba and Sparrows Point, Maryland. Little accurate information has ever been published. It contains some unusual features and is of strictly Bethlehem design.

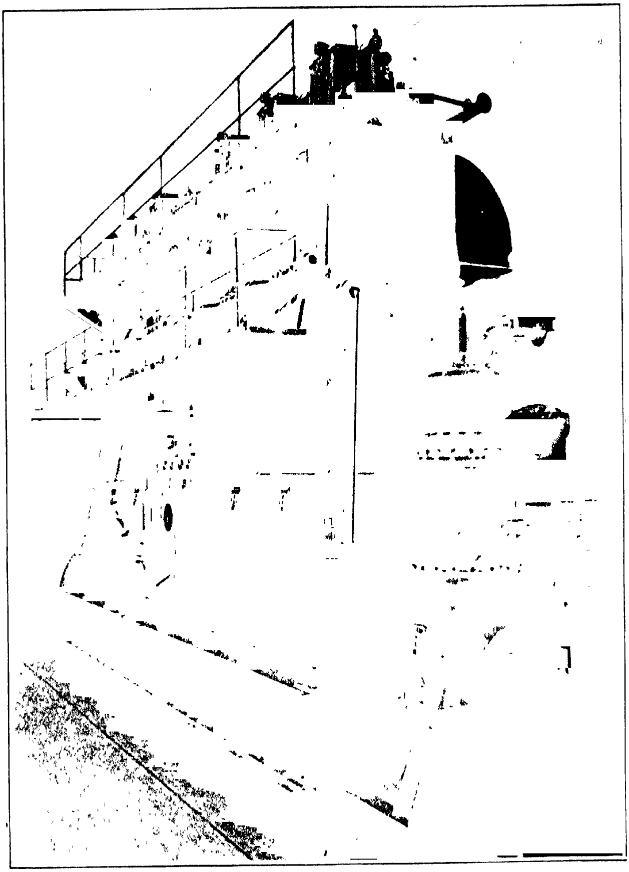


FIG 41.—Cramp's B. & W. 1,750 hp marine Diesel.

Dow Pump & Machinery Co. Marine Diesel.—The Dow Co. has designed a marine Diesel which follows in the main the lines of its stationary engine.

New York Shipbuilding Co. Marine Diesel.—A license to manufacture the Werkspoor Marine Diesel is held by the New

York Shipbuilding Co. Up to the present time two engines have been built.

Sun Shipbuilding Co. Marine Diesel.—The Sun Co. has the American rights of the opposed-piston (Doxford Sytem) engine.

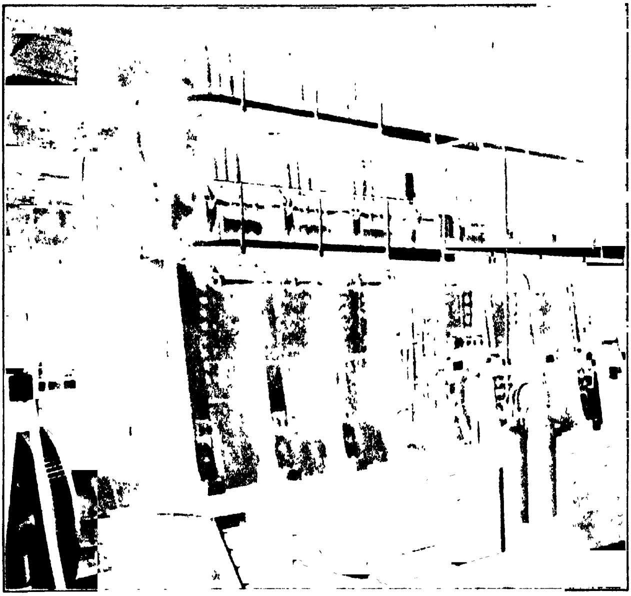


FIG 41A.—Worthington 1,750 four-stroke-cycle marine engine.

So far only development work has been carried on. The engine follows European designs but is a solid-injection Diesel.

James Craig Marine Diesel.—The Craig Co. has built marine internal combustion engines for many years. In the past a number of the Craig Diesels have been used in the United States Navy. At present the company is building a 1,750-b.hp. unit.

CHAPTER V

ECONOMIC STATUS OF THE DIESEL

General.—In considering the purchase of power-generating machinery the factors that should largely determine the choice of engines are: first cost; fuel costs; labor costs; and maintenance.

It will be found that the Diesel engine is not always the logical unit to purchase. Often steam engines, such as the uniflow or Corliss, are the logical units. Under other circumstances electric power should be purchased, while under certain plant conditions the less efficient semi-Diesel will prove to be most economical.

First Cost.—It is almost impossible to give any data on Diesel prices that will prove consistent at all times and places. Engines have been quoted as low as $45 per brake-horsepower and as high as $120. The former applies to large engines, over 750 hp., while engines of small power are quoted at the higher price.

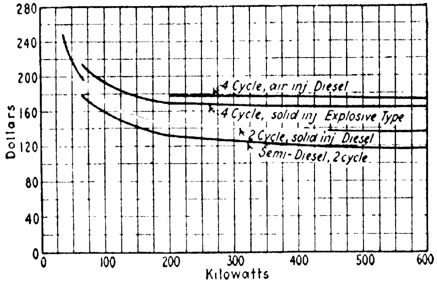


FIG. 42.—Initial cost of oil engine per k w. capacity.

As a rule, pipe lines being excepted, Diesels are direct-connected to generators. The purchaser is interested in total costs including all accessories; such as, oil tanks, pumps, etc. In Fig. 42 are shown a set of curves covering Diesel, semi-Diesel, and solid-injection engine-generator costs per kilowatt of installed capacity. It will be noted that while it was stated that the large Diesels have been quoted at $45 per horsepower and small units at $120

—the costs per kilowatt do not have the same ratio. This is due, first, to the fact that only in exceptional cases will the prices per brake horsepower fall below $65; second, as the engine sizes increase the revolutions per minute decrease and, consequently, the generator cost increases. The costs given in Fig. 42 may be used in making estimates. The initial investment for the Diesel plant is higher than for a steam plant of the average type. If economical steam-units are considered, and the extra room space required for the boilers and fuel storage be not ignored, the difference in first costs is slight and by no means unsurmountable, and will, in most cases, be more than counterbalanced by the lower operating charges of the Diesel plant.

Diesel and Semi-Diesel Compared.—As Fig. 42 shows, the first cost of a semi-Diesel plant is less than that of a Diesel. By reason of the higher operating charges of the semi-Diesel it is the logical choice only in plants that run a few months a year, as example, a cotton gin, or in plants so small that the saving in operating charges is small. Many consulting engineers claim that below 100 hp. the semi-Diesel proves more economical where the plant operates at a *capacity-use factor* of less than 70 per cent. This means that the engine operates fully loaded less than 70 per cent of the time or at 70 per cent load less than 100 per cent of the time, based on a 10 hr. day.

The semi-Diesel's fuel consumption per kilowatt-hour will exceed that of the Diesel by at least 0.22 lb. in the average plant. In addition, lubricating oil is consumed in the semi-Diesel at the rate of about 1 gal. for each 300 kw.-hr., and the Diesel should deliver, 1,500 to 2,000 kw.-hr. with the use of 1 gal. of lubrication. These two items make the semi-Diesel's operating costs run from 2 to 4 miles above that of the Diesel's. If the plant is run at a high capacity-use factor the higher operating costs overbalance the lower fixed charges.

Above 100 hp. it would seem that the capacity-use factor must be less than 50 per cent to make to semi-Diesel competitive.

The Solid-injection vs. The Diesel Engine.—By reason of the excellent fuel economy, of the solid-injection engine, the Diesel's total costs will exceed that of the former in sizes below 300 to 400 hp. There are other factors to be considered, although the determination of the comparative influence of these factors is difficult.

The solid-injection engine by reason of its cycle, being an explosion or Otto-cycle is often called upon to withstand greater maximum pressures than is the Diesel. The process of combustion is not as closely regulated. As a result of these two advantages possessed by the Diesel, the latter is classed by many as being more reliable. Below 300 hp. reliability is not as important a factor as it is in larger units.

Labor Costs.—The oil-engine plant in capacities below say 3,000 kw. can be operated with a smaller crew than can a high-grade steam plant containing stoker-fired boilers, steam turbines or uniflow engines, condensers, etc.

In a plant containing a single unit of less than 300 kw. one man per shift is ample. In addition, part of a laborer's time, say one-half, should be charged against the engine for the engineer will require assistance at times of overhauling, repair, etc. In a plant of two units of less than 750 kw. two men per shift are advisable. In plants containing two or more units there should be at least two men per shift. If over 1,000 kw. there should be an extra man during the day shift.

The pay of Diesel operators is, it is true, somewhat higher than that of steam plant engineers in the same locality. The difference is not large enough to influence the plant cost to any great extent.

It is impossible to operate a 300 kw. steam plant with one man per shift. In large plants up to 2,000 kw. the Diesel labor costs are under those of the steam plant. When plants of 5,000 to 10,000 kw. are considered it will be found that the steam plant needs but little more labor than would the oil engine plant.

The actual labor costs are given in a number of tables later in this chapter.

Since this chapter is primarily written for the manager, a word in regard to the question of operating labor is not amiss. It has been claimed too often by ambitious salesmen and uninformed engineers, both consulting and operating, that the Diesel engine requires no skilled attendant. There is no doubt that the amount of labor is much less than in a steam plant, but the Diesel engineer should be a mechanic. It is not necessary that the operator be a machinist, but he must be familiar with machinery, have some experience in caring for high-grade machines and, above all, must understand the principle of operation and be of the type that will follow instructions and will make adjustments as required.

As a general proposition the so-called "automobile mechanic" does not develop into a successful Diesel engineer. He has been trained in a shop where poor work, crude repairs, and indifferent machine work are customary. It may be stated with truth that a man who has been in charge of a steam power plant makes the best Diesel operator. He is accustomed to the handling of power machinery, understands the secret of keeping the engine in service, and is usually a fair mechanic.

Maintenance Charges.—Maintenance charges are in direct proportion to the reliability of the unit.

As a general rule all Diesel engines are very easy to operate, it merely requiring an understanding of the principles upon which they run, as well as some experience with such a type engine. It must always be kept in the best of repair if it is expected to run well at all. It should be kept in as good condition as the day when it was bought, which in turn eliminates excessive depreciation charges. A careful observance of the kind of fuel oil and lubricating oil used will lead to very little trouble with the moving parts. It is generally assumed that a Diesel engine requires more repairs than a steam plant. However, the experience with the modern types has proved that the difference is very slight and in favor of the Diesel plant.

Failures or high maintenance charges of early oil engines may be traced in most instances to lack of experienced operators. As soon as engineers became acquainted with the Diesel peculiarities, repairs and shut downs decreased.

Fuel Consumption.—It will be found that the several manufacturers of Diesel engines offer practically the same guarantees as to fuel consumption per brake-horsepower hour. For the same cylinder sizes the four-stroke-cycle engine will show a consumption a trifle lower than will the two-stroke-cycle. This is due to the more complete scavenging of the four-stroke-cycle cylinder of the burnt gases; to offset this, the two-stroke-cycle engine should have a lower first cost per horsepower. This usually equalizes the total cost per horsepower-hour.

Purchasers should not be too much impressed with astonishing claims made by some particular manufacturer. Under ordinary circumstances the fuel consumption of all Diesels, regardless of the size, will not vary over from 5 to 7½ per cent. It is possible under exceptional shop conditions to obtain a record impossible in ordinary operation in the field.

Many cannot understand how the European engines are able to show records below those of American units of like powers. The difference is more apparent than real. Often it will be found that the injection air was supplied by a separately-driven compressor. This will make a difference of, say, 8 per cent.

In Table V are given the usual guarantees made by American builders. In plant operation these are often lowered by as much as 5 per cent.

Table V.—Manufacturers' Fuel Guarantees

Make	Type	Size	Full load, lb.	¾	½	¼
National Transit	H. A.	65–300	0.48	0.50	0.58	
De La Vergne	H. A.	100–300	0.42	0.435	0.52	
Korting...	H. A.	300	0.42	0.43	0.52	
Snow...	H. A.	65–600	0.50	0.52	0.60	
McEwen	H. A.	65–300	0.48	0.50	0.58	
Nelseco..	V. M. A.	120–600	0.45			
Fulton	V. A.	120–750	0.41	0.425	0.45	0.625
Busch-Sulzer... ...	V. A.	500	0.42	0.43	0.52	
McIntosh & Seymour	V. A.	500–1,750	0.42	0.43	0.52	
McIntosh & Seymour	V. M. A.	600	0.404			
Standard Fuel Oil Engine.	V and H. B.	60–300	0.48	0.50	0.56	0.84

H = horizontal engines.
V = vertical engines.
M = marine engines.
A = four-cycle.
B = two-cycle.

Standard Diesel Fuel Consumption.—In Fig. 43 are shown results of a series of tests on a 100 hp. two-cylinder Standard two-stroke-cycle vertical engine at various speeds. The lowest fuel consumption, 0.455 lb. per brake horse-power-hour, was at a speed of 225 r.p.m. At higher speeds the poorer scavenging reduced the engine efficiency.

In Fig. 44 are given results of a test on a horizontal Standard two-stroke-cycle engine. The fuel consumption is somewhat higher than that obtained with the vertical engine. The difference is due both to the more improved design of the vertical engine and also to the better test conditions.

The fuel consumption to be expected where the engine is in good working condition is shown in Fig. 44 *A*. The table gives the fuel cost per horsepower per year.

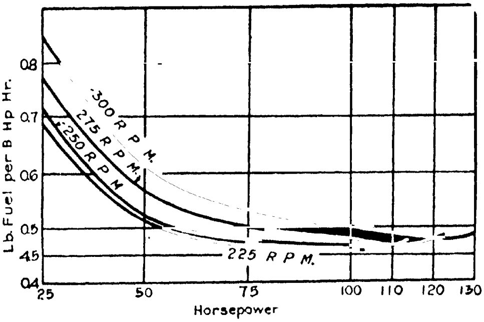


FIG. 43.—Fuel consumption of 100 hp standard engine at various speeds.

Snow Diesel Fuel Consumption.—Figure 45 is the result of a test on a 300-hp. Snow Diesel. It will be noted that the test gives fuel consumptions that are 20 per cent lower than this company's standard guarantees.

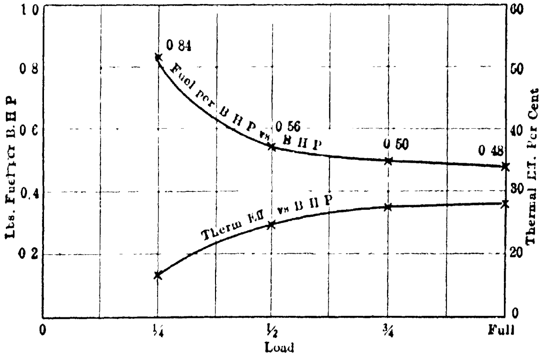


FIG. 44.—Test on horizontal standard Diesel.

McEwen Diesel Fuel Consumption.—Figure 46 shows a test on a 65-hp. McEwen Diesel. The fuel consumption at full load is remarkably low. The increase at three-quarters and half-load over the full-load consumption is quite marked. From this it would appear that the engine's rating was not as high as it should be. In other words, the rating should be raised, making the engine a 75-hp. unit.

ANNUAL FUEL COST OF ONE BRAKE HORSEPOWER

HOURS OF USE PER DAY	COST OF FUEL OIL IN CENTS PER GALLON														
	2½	3	3½	4	4½	5	5½	6	6½	7	8	9	10	11	12
	COST OF FUEL FOR ONE BRAKE HORSEPOWER ONE YEAR (300 DAYS)														
1	$0.45	$0 54	$0 63	$0 72	$0 81	$0 90	$0 99	$1 08	$1 17	$1 26	$1 44	$1 62	$1 80	$1 98	$2 16
2	.90	1 08	1 26	1 44	1 62	1 80	1 98	2 16	2 34	2 52	2 88	3 24	3 60	3 96	4 32
3	1 35	1 62	1 89	2 16	2 43	2 70	2 97	3 24	3 51	3 78	4 32	4 86	5 40	5 94	6 48
4	1 80	2 16	2 52	2 88	3 24	3 60	3 96	4 32	4 68	5 04	5 76	6 48	7 20	7 92	8 64
5	2 25	2 70	3 15	3 60	4 05	4 50	4 95	5 40	5 85	6 30	7 20	8 10	9 00	9 90	10 80
6	2 70	3.24	3 78	4 32	4 86	5 40	5 94	6 48	7 02	7.56	8 64	9 72	10 80	11 88	12 96
7	3 15	3 78	4 41	5 04	5 67	6 30	6 93	7 56	8 19	8 82	10 08	11 34	12 60	13 86	15 12
8	3 60	4 32	5 04	5 76	6 48	7 20	7 92	8 64	9 36	10 08	11 52	12 96	14 40	15 84	17 28
9	4.05	4 86	5 67	6 48	7 29	8 10	8 91	9 72	10 53	11 34	12 96	14 58	16 20	17 82	19 44
10	4 50	5 40	6 30	7 20	8 10	9 00	9 90	10 80	11 70	12 60	14 40	16 20	18 00	19 80	21 60
11	4 95	5 94	6 93	7 92	8 91	9 90	10 89	11 88	12 87	13 86	15 84	17 82	19 80	21 78	23 76
12	5 40	6 48	7 56	8 64	9 72	10 80	11 88	12.96	14 04	15 12	17 28	19 44	21 60	23 76	25 92
13	5 85	7 02	8 19	9 36	10 53	11 70	12 87	14 04	15 21	16 38	18 72	21 06	23 40	25 74	28 08
14	6 30	7 56	8 82	10 08	11 34	12 60	13 86	15 12	16 38	17 64	20 16	22 68	25 20	27 72	30 24
15	6 75	8 10	9 45	10 80	12 15	13 50	14 85	16 20	17 55	18 90	21 60	24 30	27 00	29 70	32 40
16	7.20	8 64	10.08	11 52	12 96	14 40	15 84	17 28	18 72	20 16	23 04	25 92	28 80	31 68	34 56
17	7.65	9 18	10 71	12 24	13 77	15 30	16 83	18 36	19 89	21 42	24 48	27 54	30 60	33 00	36.72
18	8 10	9 72	11 34	12 96	14 58	16 20	17 82	19 44	21 06	22 68	25 92	29 10	32 40	35.64	38 88
19	8 55	10 26	11 97	13 68	15 39	17 10	18 81	20 52	22 23	23 94	27 36	30 78	34 20	37 62	41 04
20	9 00	10 80	12 60	14.40	16 20	18 00	19 80	21 60	23 40	25 20	28 80	32 40	36 00	39 00	43 20
21	9 45	11 34	13 23	15 12	17 01	18 90	20 79	22 68	24 57	26 46	30 24	34 02	37 80	41 58	45 36
22	9 90	11 88	13 86	15 84	17 82	19 80	21 78	23 76	25 74	27 72	31 68	35 64	39 60	43 56	47 52
23	10 35	12 42	14 49	16 56	18 63	20 70	22 77	24 84	26 91	28 98	33 12	37 26	41 40	45 54	49 68
24	10 80	12 96	15.12	17 28	19 44	21 60	23 76	25 92	28 08	30 24	34 56	38 88	43 20	47 52	51 84

FIG. 44A.—Usual fuel economy.

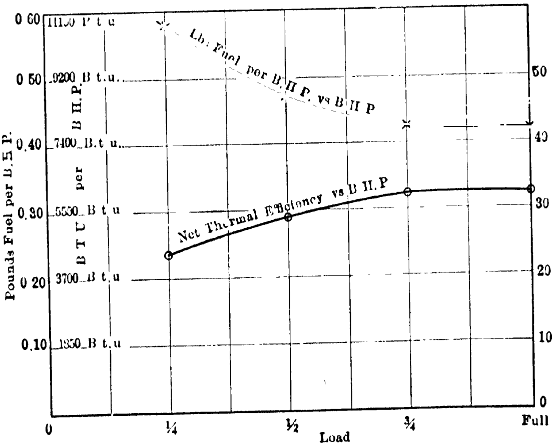


FIG. 45.—Test results of a Snow Diesel.

Busch-Sulzer Diesel Fuel Consumption.—Figure 47 shows the results of a test on a 500-hp. Busch-Sulzer Type B Diesel. This test was run without any preparation in the way of engine adjustments and represents actual operating results.

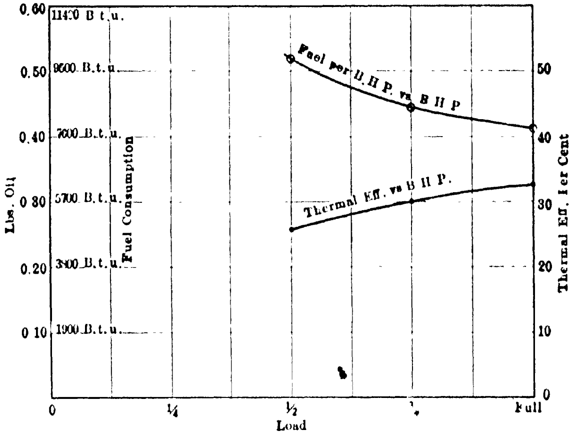


Fig. 46.—Test on a 65 hp. McEwen engine.

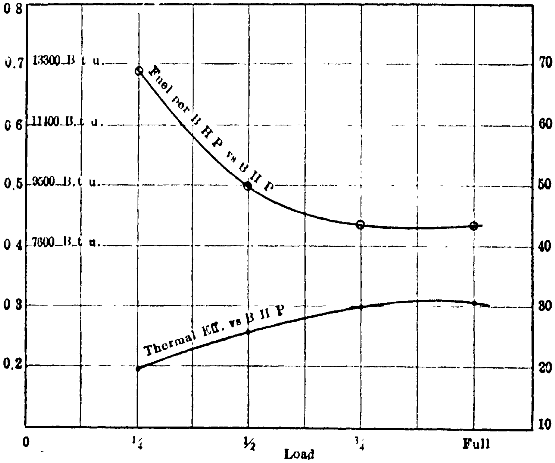


Fig. 47.—Test on a 500 hp. Type *B* Busch-Sulzer Diesel, cylinders 16 × 24, speed 200 r.p.m.

McIntosh & Seymour Diesel Fuel Consumption.—Table VI outlines complete tests on three 500-hp. McIntosh & Seymour Diesels direct connected to G. E. alternators. These tests were

TABLE VI.—SUMMARY OF TESTS ON PARIS (TEX.) DIESEL ENGINES

	Unit No 1				Unit No 2				Unit No 3			
	1	3	2	4	5	6	7	8	9	10	11	12
Approximate load	Full	Three-fourths	Half	Quartre	Full	Three-fourths	Half	Quarter	Full	Three-fourths	Half	Quarter
Kw. (average)	340.6	253.3	168.9	86.1	333.4	250.5	153.4	82.4	340.0	249.7	162.9	79.6
Load factor (kw.), per cent	97.5	72.5	48.2	24.6	95.2	71.6	44.2	23.5	97.2	71.3	46.5	22.7
Kv.-a (average)	400.7	301.5	211.1	117.9	396.2	316.5	199.1	112.8	409.5	320.1	211.5	109.0
Load factor (kv.-a), per cent	91.7	69.0	48.2	27.0	90.8	72.5	45.6	28.0	93.7	73.3	48.4	25.0
B.hp. (average)	481.3	359.8	240.2	127.1	472.6	355.6	222.0	117.3	482.0	356.9	234.3	117.2
Kw.-hr. generated (total)	1,299.3	891.1	577.0	101.9	1,103	784	413.1	49.45	1,008.5	919.7	450.9	55.7
Time start, p.m. and a.m	2·29	7.13	2.53	11.37	6:00	2:21	10.01	1.16	5·55	2.02	9.41	12·54
Time stop, p.m. and a.m	6·18	10.49	6.18	12.48	9·18½	5:28½	12.42½	1.52	8.53	5·43	12:27	1:36
Duration of test, hours	3.816	3.517	3.417	1.183	0.84	3.125	2.692	0.6	2.966	3.683	2.767	0.7
Power factor (average)	0.85	0.84	0.80	0.73	3.308	0.79	0.77	0.73	0.84	0.78	0.77	0.73
Kw. for excitation (average)	8.1	6.91	7.64	7.42	7.25	7.90	6.50	7.17	7.4	7.0	7.1	7.3
Fuel oil consumed (gal.)	111	77	56	14	96	68	41.5	6.7	84	77½	44	7
Fuel oil consumed (lb.)	783	543	396	98½	673	481½	290½	47	592½	542½	310	49
Fuel oil consumed (B t u)	15,321,744	10,621,080	7,789,616	1,923,803	13,145,036	9,396,954	5,657,778	918,615	11,590,485	10,625,948	6,064,220	955,500
Fuel oil per kw.-hr. (lb.)	0.603	0.609	0.686	0.966	0.610	0.614	0.703	0.950	0.590	0.590	0.688	0.880
B.t.u. per kw.-hr	11,792	11,930	13,500	18,879	11,917	11,984	13,720	18,576	11,492	11,553	13,448	17,154
B.t.u. per b.hp. hr	8,342	8,396	9,491	12,795	8,407	8,450	9,467	13,052	8,037	8,083	9,354	11,646
Generator efficiency, per cent	92.6	91.8	89.5	83.0	92.5	91.4	88.7	86.0	92.6	91.3	89.2	82.8
Thermal efficiency, per cent	28.9	28.6	25.3	18.0	28.7	28.5	25.0	18.4	29.7	29.4	25.4	19.9
Water for cooling (total gal.)	14,653	11,606	....	4,436	15,382	11,250	9,368	....	14,949	13,038	9,629	
Water per minute (gal)	64	55	....	62½	77.5	60.0	58	. .	84	59	58	
Number units in operation	3	3	3	2	3	3	3	2	3	3	3 & 2	
Station load (average kw.)	635	697	660	370	703	688	478	385	666.6	520	483	
Station factor (average)	60.5	64.2	62.8	35.2	67	65.5	45.5	35.6	63.5	49.5	46.0	

carried out while the engines were in actual service, delivering current into a 100-kv. high-tension line. The efficiencies of the three units were practically uniform and represents values that are about as high as can be expected in units of this size.

Actual Operating Results.—The preceding figures represent test values at certain stated loads. It is impossible to secure a given load factor at all times. Consequently the fuel consumption of a Diesel in actual operation will not check with test results. In a few instances, such as with the plant containing the three 500-hp. engines appearing in Table VI, the plant can operate at practically full load at all times. The fuel consumption should then check with test results. In the Paris, Texas, plant the test gave a fuel consumption at full load of 0.603 lb. per kilowatt-hour. The actual operating consumption in this plant is 0.672 lb. per kilowatt-hour while the load ranges from three-quarters to full load. This is but 10 per cent above the test results.

Unfortunately, such ideal loads seldom are encountered. The usual Diesel engine carries a load from one-quarter to full load, and the fuel consumption is considerably different from full-load test results.

Table VII is a compilation of the hourly loads in a plant containing three old 225-hp. Diesels and a 500-hp. Busch-Sulzer. Type B Diesel. The 500-hp. engine has an individual fuel tank while all three of the older engines obtain their supply from a common tank. Hourly readings of the fuel consumed, actual kilowatts delivered to the switchboard and the indicated kilowatt load are entered in a station log. . It will be noted that the 500-hp. unit operated 16 hr. daily and carried a load ranging from one-third to one-half load. One hour's fuel consumption (from 9 a.m. to 10 a.m.) is evidently erroneous, but this was rectified in the subsequent hourly readings. The average consumption of 0.71 lb. per kilowatt-hour at half load is exceptionally good and is evidence of the high-class attendance employed in the plant. The interesting figures in this log are the fuel consumptions of the obsolete 225-hp. Diesels. These engines are from 10 to 15 years old and have had hard service. Nevertheless, the fuel consumption of these engines at less than half load averages -0.909 lb. per kilowatt-hour. Although the design and workmanship of these early American Diesels are often ridiculed, it is questionable if the more modern engines will show any better results after the same length of service.

TABLE VII.—DAILY LOG, DIESEL PLANT A

Time	500 h.p.		225 h.p.		225 h.p.		225 h.p.		500 h p. fuel		Fuel, other engines	
	Ind. kw.	Kw.-hr.	Ind. kw.	Kw.-hr.	Ind. kw.	Kw.-hr.	Ind. kw.	Kw.-hr.	Total, gals.	Lb. per kw.-hr.	Total, gals.	Lb. per kw.-hr.
12 P.M.	On 8.35 A.M.		100	...	80	...	100					
1 A.M.			90	70	80	70	80	70	...	....	25	0.87
2			90	75	70	65	80	70	...	....	28	0.97
3			95	85	65	75	80	80	...	....	28	0.85
4			90	70	75	70	80	70	...	....	26	0.90
5			90	75	70	65	80	70	...	....	27	0.91
6			80	80	70	80	70	80	...	....	34	1.18
7			130	75	..	60	130	75	...	....	28	0 97
8			130	120	..	...	130	120	...	....	24	0.73
9	240	70	...	75	..	...	...	65	9	0.94	18	0 965
10	230	240	Off 8:40 A.M.		Off 6:55 A.M.		Off 8 35 A.M.		18	0.55		
11	230	210							21	0.73		
12 Noon	210	210							20	0 70		
1 P.M.	200	210							21	0 73		
2	200	180							18	0.73		
3	230	210							21	0 73		
4	250	215							21	0 71		
5	250	210							20	0 70		
6	200	170							20	0 86		
7	200	180							20	0 81		
8	260	240							22	0 67		
9	300	260							23	0 65		
10	240	260							23	0 65		
11	220	210							22	0 76		
12 P.M.	220	210							21	0 73		
Total kw.-hr.	...	3285	...	725	...	485	...	700	320	....	238	.....
Fuel per kw.-hr., average	...	...	...	...	...	...	...	...	...	0 71	...	0 909

Operation record—Plant A.
Units: 500-h.p. Busch-Sulzer Type B.
225-h.p. Busch-Sulzer Type A (old Adolphus Busch engines).

Power Station at Plant B for Twenty-four Hours Ending Midnight

	Phase Voltage			Kw-hr record meter	Gen 1, 350 kw.cap.				Gen. 2, 150 kw cap.				Gen. 3, 100 kw.cap				Gen. 4, 150 kw.cap				Diesel fuel tank			Load factor, per cent.	Lbs. fuel per kw.-hr.
					Amperes			Ind. kw	Amperes			Ind kw.	Amperes			Ind kw	Amperes			Ind kw	Gals. used per hr		Total used		
	A	B	C		A′	B′	C′		A′	B′	C′		A′	B′	C′		A′	B′	C′		350 kw. unit	Other units			
12 P.M.	121	122	120	....	..	..	..	...	35	36	36	100	27	27	27	80	25	24	24	100	Gals.	Gals.	Gals.	..	.....
1 A.M.	121	122	120	210	..	..	..	...	30	30	30	90	27	28	27	80	26	27	26	80	..	25	25	47	0 869
2	121	121	120	210	..	..	..	...	31	30	31	90	25	26	25	70	26	26	26	80	..	28	28	47	0.973
3	121	121	120	190	..	..	..	...	32	31	31	95	26	25	26	65	25	25	26	80	..	28	28	42	1.075
4	121	121	120	185	..	..	..	...	31	30	31	90	26	27	26	76	24	25	25	80	..	26	26	41	1.026
5	121	121	121	188	..	..	..	...	30	30	30	90	26	27	27	70	23	22	23	80	..	27	27	41	1.050
6	121	121	120	190	..	.	.	...	28	30	28	80	Off at 5·55				22	23	22	70	..	34	34	42	1.305
7	121	121	120	110	..	..	.	...	34	35	35	130	.	.	.	...	32	33	33	130	..	20	20	37	1.327
8	121	121	120	170	On at 8 30				33	33	35	130	.	..	..	...	31	31	31	130	..	24	24	57	1.030
9	121	121	120	180	65	65	69	240	Off at 8 40					.	.	...	Off at 8.35				.	18	18	50	0.730
10	121	121	120	180	63	65	69	230	.	.	.	...	.	.	..	...	.	.	.	..	9	10	19	50	0 770
11	121	120	120	175	60	63	65	230	.	.	.	...	.	.	..	..	.	.	.	..	18	..	18	50	0.768
12 Noon	121	121	120	180	60	62	63	210	..	.	..	..	.	..	..	..	.			..	20	..	20	50	0.810
1 P.M.	121	121	120	100	60	60	63	200		.	..	...	.		.	. .		.	.	...	20	..	20	28	1.460
2	121	121	120	160	60	60	63	200	.	.	.	...	.	.	..	...	.	.	..	...	21	..	21	46	0 957
3	121	121	120	180	60	62	63	210	.	.	..	..	..	.	.	..	.	.		..	18	..	18	50	0 730
4	121	121	120	175	60	62	63	210	..	..	.	...	..	..	..	...	.	..	..	...	21	..	21	50	0.876
5	121	121	120	175	60	60	60	200	.	..		.	.	..	..	.		.	.	...	21	..	21	50	0 876
6	121	121	120	175	63	65	69	240	On at 6 05				.	.		.		.	.	.	21	..	21	50	0 876
7	121	120	120	180	63	65	67	240	28	30	28	80	.	..		..	.		.	..	20	10	31	36	1.260
8	121	121	120	225	63	60	65	230	34	35	34	130	.	.	.	...		.	.	..	21	12	33	45	1.070
9	121	120	120	255	60	60	63	200	34	35	35	130	.	..	.	...	.		.	.	21	12	33	51	0.945
10	121	121	121	250	60	60	63	200	33	35	33	130	..	.	.	..	.	.	.	...	20	12	32	50	0.930
11	120	120	121	130	60	60	63	200	Off at 10:00				.	.	.	.	.	.	..	. .	20	..	20	37	1.123
12 P.M.	120	120	121	175	60	60	62	200	.	..	.	. .	.	.		..	.	.	..		21	..	21	50	0 875
Total.......	...	...	...	4348	.	.	.	..	..	..	..	...	.	.		...		.	.	.	.	.	579	..	0.972

Table VIII is the summary of a daily log of a plant having two 225-hp. and one 170-hp. old Diesels and one 500-hp. McIntosh & Seymour Diesel. The average fuel consumption was 0.972 lb. per kilowatt-hour. This high value is due to the conditions of the old engines and to the engineer's unwillingness to have the 500-hp. engine carry the night load even though it was less than the engine rating. Frequently the engineer is blameable for a higher fuel consumption since he is unwilling to work the engines at their economical factors.

An interesting feature in connection with this plant is that one of the 225-hp. engines was overhauled in 1921. After reboring the cylinders, purchasing new pistons, valves, etc., a test was run on the engine. At about the same time tests were run on the 500-hp. Busch-Sulzer Type B which had been in constant service for 5 years without overhauling, and upon a 500-hp. McIntosh & Seymour installed in 1917. The results of these tests are shown in Table IX. Here the 500-hp. McIntosh & Seymour unit is marked as *B*, the Busch-Sulzer engine, as *A*, and the 225-hp. engine as *C*. The reader's attention is called to the high efficiency of the 225-hp. engine after overhauling. Its fuel consumption of 0.74 lb. per kilowatt-hour after 10 years of service is practically the same as that of the more modern engines.

TABLE IX.—RESULTS OF TESTS ON THREE DIESEL ENGINES

Engine	*A*	*B*	*C*	
Horsepower	500	500	225	
Year manufactured	1916	1917	1910	
Year tested	1921	1921	1921	1911
Temperature cooling-water inlet	80	82	80	
Temperature cooling-water outlet	126	130	126	
Oil consumed, pounds	20,556	21,071	11,970	
Kilowatt-hour output	30,970	29,050	16,090	
Length of test, hours	100	100	100	1
Load factor, per cent	86	90	90	100
Pounds fuel per kilowatt-hour	0.66	0.72	0.74	0 69
Kilowatt-hours per barrel fuel	476	436	425	455
B.t.u. per kilowatt-hour	12,478	13,636	13,994	13,045
Fuel cost per kilowatt-hour, mills ($3.23 per barrel)	6 60	7.41	7.10	

Fuel Supply.—In considering the choice of a prime mover for a power plant, the investigator is frequently prompted to install other than oil engines for fear of oil scarcity. Much has been written and spoken concerning the danger of the total exhaustion of the petroleum supply. While economy should be practiced in respect to the oil-field supply, no immediate danger of exhaustion exists. At the present time there is strong indication that Canada contains a very important oil field. In Mexico, South America, and the West Indies there are petroleum supplies that are still untouched. There is no doubt that petroleum prices in the future will advance with increased consumption and rising coal prices. It is hardly probable that fuel oil will ever reach a price that would make the Diesel fuel costs higher than the costs of a steam plant in favorable localities. Recently crude oil has been selling at one-half the 1920 quotations, and fuel oil is obtainable at from 85 cents to $1.50 per barrel in the Middle West. At these prices the Diesel economy cannot be attained by any of its competitors. In addition, the vast beds of shale in the West offer an oil supply for 1,500 years at the present consumption of 500,000,000 barrels of crude oil per year.

The foregoing discussion has been incorporated in this volume for the benefit of those engineers who are interested in the economic problem of power generation and of those managers and owners of factories, etc. who are seeking for a more economical power than is at present used. Every power engineer who is worthy to be called an engineer is or should be interested in the development of new processes and new machinery.

Production Costs.—While the Diesel is superior to any other form of prime mover in thermal efficiency, the actual commercial efficiency of the engine may be even less than a steam turbine at low loads. It is obvious that the Diesel plant investment is high with consequent high interest and depreciation (fixed) charges. If the plant is operated at a low load factor, the fixed charges and the labor charge will cause the total cost per kilowatt-hour to be excessive. Figure 48 is a chart which shows the total cost per kilowatt-hour in a graphic manner. This plant had a total installed capacity of 500 kw. The entire plant cost $81,600. Table X gives the various charges used in developing the curves in Fig. 48. It is apparent that the two factors that largely determine the total cost per kilowatt-hour at any load are the labor and fixed charges. These two items, divided into hourly charges, are here assessed against the hourly load and cause the total cost per kilowatt-hour to mount very rapidly on low loads. The solution of economical Diesel operation is a high load factor which would tend to reduce the overhead and labor charges. ·This particular

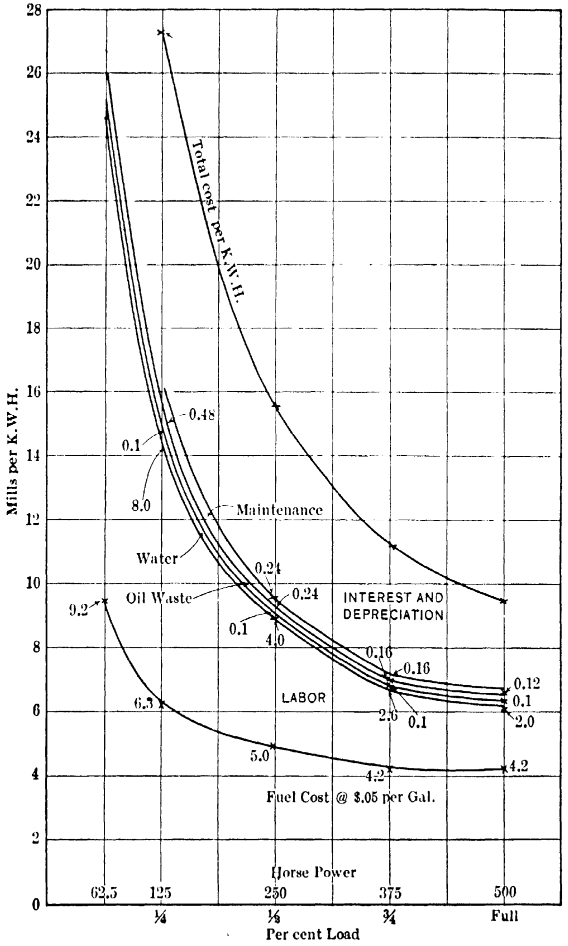


Fig. 48.—Influence of load on total cost per kw.-hr.

plant employs four men, and the capacity could be doubled without any increase in the labor cost, although the fixed charge would increase in ratio with the engine capacity. These curves

TABLE X.—PRODUCTION COSTS, 500-HP. DIESEL PLANT
In Mills per Kilowatt-hour

Load	Interest and depreciation	Fuel	Labor	Water	Oil and waste	Maintenance	Total
Full	3 0	4 2	2 0	0 1	0 12	0 12	9 54
¾	4 0	4 2	2 66	0 1	0 16	0 16	11 28
½	6 0	5 0	4 0	0 1	0 24	0 24	15 58
¼	12 0	6 3	8 0	0 1	0 18	0 18	27 36

Fuel at 5 cents per gallon.

Labor:

Chief engineer at $200 per month.
Two watch engineers at $150 per month
Two helpers at $80 per month

Overhead:

7 per cent interest
8 per cent depreciation
—
15 per cent total.

TABLE XI—PRODUCTION COST, DIESEL PLANT AT PLANT C, SEPT. 2, 1917

Daily plant operation report ... Co	To-day	Month to date
Kw.-hr. output	1,930	4,810
Peak load kw. at 9 P.M.	150	200
Peak load factor per cent.	57	50
Station load factor per cent.	11	14
Generator load factor per cent.	38	29
Fuel used (gals. oil) Diesel fuel oil	340	762
Fuel used per kw.-hr. output, gals. oil	0.18	0 16
B.t.u. per kw.-hr. output	22,420	.....
Fuel cost—total (cents)	884.0	1,981 0
Fuel cost (mills) per kw.-hr. output	4 6	4 1
Labor cost—total (cents)	1,370.0	2,740 0
Labor cost (mills) per kw.-hr. output	7 2	5 7
Miscellaneous cost—total (cents)	500 0	1,000.0
Miscellaneous cost (mills) per kw.-hr. output	2 6	2.1
Maintenance cost—total (cents)	700 0	1,400 0
Maintenance cost (mills) per kw.-hr. output	3.7	3.0
Production cost—total (cents)	3,454 0	7,121.0
Production cost (mills) per kw.-hr. output	18.0	15 0
Time generators in service—gen. hr.	32	67
Temperature (deg. F.) circulating water	76	76
Signed		

prove that Diesel plants must be fairly well loaded to be economical. For this reason it frequently pays a plant to give a very low power rate in order to increase the load factor.

The engine should be of a size appropriate to the load, and with an increase in load a second unit may be installed.

Table XI is the daily report of a Diesel plant having a 350-kw., two 125-kw. and one 100-kw. Diesel units. The smaller units are very old and have a high fuel consumption. The load factor is very low which results in an excessive fuel charge even on the modern 500-hp. unit. The old units required much overhauling—this shows up in the maintenance charge. The total production cost was 18 mills per kilowatt-hour while the fixed charges are about 10 mills per kilowatt-hour, based on the daily output. However, this total cost of 28 mills per kilowatt-hour compares very favorably with steam plants having similar load conditions.

TABLE XII.—ACTUAL UNIT PRODUCTION COSTS, PARIS AND TYLER DIESEL STATIONS, SEPT. 1 TO DEC. 31, 1916

Article	Paris	Tyler
Data:		
Station output (m. kw.-hr.)	1,565	499
Rating of plant (kw.)	1,050	600
Station factor, per cent	51	28½
Total fuel oil (gals.)	149,072	78,455
Pounds oil per kw.-hr. output	0.672	1.100
B.t.u. per kw.-hr. output	13,100	21,400
Production costs (mills per kw.-hr.):		
All labor	1 44	2.24
Fuel oil	3 07	5 18
Water	0 09	0 19
Lubricants and waste	0.04	0 56
Miscellaneous supplies and expense	0.10	0.29
Maintenance of engines	0.04	4 48
Maintenance of buildings	0.05	0.05
All other maintenance	0.15	0.61
Total production cost, mills	4.98	13.60

Oil at 3 cents per gal.

Table XII covers the results of three months' operation of two Diesel plants of the Texas Light and Power Co. The Paris plant possesses three modern 500-hp. Diesels, while the Tyler plant has two double-engine units. These latter were second-hand units that were originally installed in an Eastern industrial plant. The condition of these units is revealed in the maintenance charge of 4.48 mills per kilowatt-hour. The fuel cost of the Tyler plant is 30 per cent greater than in the Paris plant. This cannot be attributed entirely to the mechanical condition of the engines but to the load factor as well. The fuel consumption of these old American Diesels is less than 10 per cent over the fuel consumption of the modern Paris engines on the same load. From this the reader can rightly conclude that the efficiency of the Diesel is fairly constant, regardless of the mechanical condition of the units, as long as the engine will function.

The engines in the Paris, Texas plant deliver power into a 100-kv. transmission line in parallel with a steam turbine plant at Waco, Texas, over a hundred miles away. It is a matter of interest that the Diesel production costs are less than those of the steam turbine plant.

TABLE XIII.—PRODUCTION COSTS, PARIS, TEXAS, DIESEL STATION

	1919	Sept 1–Dec 31, 1916
Engine hours	24,047	
Gross kw.-hr	7,580,130	
Net kw.-hr.	7,507,940	1,565,000
Station factor	93	51
Fuel oil, lb.	4,988,120	149,072
Fuel oil per kw.-hr , lb	0 657	0.672
B.t.u. per kw.-hr.	12,700	13,100
Labor per kw.-hr., mills	1 577	1 44
Fuel per kw.-hr , mills	3 130	3 07
Water per kw.-hr., mills	0 0503	0 09
Lubrication per kw.-hr., mills	0 1748	0 04
Miscellaneous supplies per kw.-hr., mills.	0 235	0 10
Building repairs per kw.-hr., mills	0 0945	0 05
Engine repairs per kw.-hr., mills	0 51	0 04
Auxiliary repairs per kw.-hr., mills	0 0103	0 15
Electrical equipment repairs per kw.-hr., mills	0 0873	
Miscellaneous station repairs per kw.-hr., mills	0 0167	
Total production cost, mills	5 99	4 98
Overhead per kw.-hr., mills.	4 85	4 85
Total cost per kw.-hr., mills.	10 84	9.83

In Table XIII are shown the production costs of this plant for three months in 1916 and for the entire year of 1919.

It will be noticed that the fuel consumption per kilowatt-hour has decreased. This is due to the increased station factor. The increase in the labor charge follows the rise in wages experienced during 1919. All the several maintenance items show an increase. With the entire equipment requiring more repairs, as is but natural, and with increasing costs, these items are very low. The record of 24,047 engine hours in 1919 shows that the three engines were in operation for over 90 per cent of the time, which compares favorably with the performance of a steam plant of like size.

If the overhead charges are assumed to remain at the same figures as were used in estimating the total power costs in 1916—namely, 4.85 mills—the present total cost per kilowatt-hour is 10.84 mills, or an increase of 1 mill per kilowatt-hour over the 1916 costs.

The operating procedure has had an important influence over the fuel and repair costs. The engines are usually run slightly below the normal horsepower rating. On Sunday one unit is shut down and necessary adjustments are made. At this time, the valves from one cylinder are removed, an extra set replacing them; the valves are cleaned during the week. Each set of valves then, is removed and cleaned every three months, while bearing adjustments, etc., are made every three weeks. A unit is out of service from 2 to 10 hours at this adjustment period.

The station force is much smaller than for a steam plant of like capacity. There is a chief engineer in charge of the plant, and on each shift there are one watch engineer, one helper and a switchboard operator, or ten in all.

Miami, Okla. Central Station.—In 1919 this Oklahoma municipality purchased a 500-hp. Fulton Diesel direct-connected to a Westinghouse generator for the light and pumping station. The single engine carried the load for over a year before a second unit was installed, and in this time there was 24-hour service with only a brief shutdown on Sunday mornings for inspection. During this time the costs were as shown in Table XIV, while Table XV gives the income and expense of the system.

Comparative Diesel and Steam Turbine Costs.—It is unfortunate that more information is not available as to costs of steam turbine plants and Diesel plants when operating under like

TABLE XIV.—POWER PLANT COSTS PER KILOWATT-HOUR, MAY 1-30, 1920

Kw.-hr. output	178,000
Fuel oil, gal	14,003
Fuel cost per kw.-hr ($2.55 per barrel)	$0 00483
Labor cost per kw.-hr	0 00518
Lubricating oil, etc	0 00580
Insurance and taxes, 4 per cent	0 00170
Depreciation, 6 per cent	0 00250
Interest, 6 per cent	0 00250
Total cost per kw.-hr	$0 02251

TABLE XV.—INCOME AND EXPENSE OF LIGHT SYSTEM

Total investment, water and light	$330,080 76
Depreciation, 5 per cent	1,375 00
Interest on bonds, 5 per cent	1,375 00
Labor	923 02
Maintenance	30 60
Lubricating oil, supplies, etc	1,040 42
Total charges	$4,744 04
Income:	
Commercial lighting	$3,584 47
Power	484 81
Water collections	2,105 46
130 street lights at $2 50	525 00
89 fire hydrants at $4	356 00
City Hall lights	9 60
Total income	$7,065 34
Net earnings	2,321 30

conditions. In Table XVI are given the comparative power-plant operating costs of a 600-kva. steam turbine and a 365-hp. Diesel engine. These two units are installed in an industrial power plant, receiving the same attention and operating under like conditions. Both units are driving generators, enabling the output of each to be accurately determined. The steam for the turbine is supplied by water-tube boilers using oil as fuel. Since the boiler efficiency is generally higher with oil than with coal, the B.t.u. per kilowatt-hour is lower than the result obtained with coal.

The Diesel exhaust line is equipped with a heater that heats the boiler-feed water and the Diesel should have been credited with the heat so added. The turbine circulating-water and air pumps are motor-driven, and no deduction has been made for this power. In all respects the turbine has been favored.

TABLE XVI.—COMPARATIVE OPERATING COSTS OF DIESEL AND STEAM TURBINE

Week (132 hr.)	Turbine				
	1	2	3	4	5
Kw.-hr. output	58,210	66,050	42,359	64,025	74,925
Fuel oil, gal.	23,455	25,950	19,389	24,474	26,836
Kw.-hr. per gal.	2.48	2.55	2.23	2.61	2.79
B.t.u. per kw.-hr	51,320	50,000	57,200	48,800	45,300
Fuel per kw.-hr., mills	14.1	13.7	15.7	13.4	12.9
Lubricating oil per kw.-hr., mills	0.1	0.1	0.2	0.15	0.15
Labor per kw.-hr., mills	2.4	2.2	3.4	2.2	2.00
Operating cost per kw.-hr., mills	18.6	16.0	19.3	17.75	15.05

Week (132 hr.)	Diesel				
	1	2	3	4	5
Kw.-hr. output	10,590	25,725	15,680	21,215	24,240
Fuel oil, gal.	1,026	2,413	1,524	2,047	2,363
Kw.-hr. per gal.	10.3	10.6	10.2	10.3	10.2
B.t.u. per kw.-hr.	12,370	12,000	12,300	123,700	12,300
Fuel per kw.-hr., mills.	3.4	3.3	3.4	3.4	3.4
Lubricating oil per kr.-hr., mills	0.2	0.1	0.15	0.1	0.11
Labor per kw.-hr., mills	13.0	5.6	9.30	6.6	5.6
Operating cost per kw.-hr., mills	16.6	9.0	12.85	10.1	9.11

In the figures given, the labor per kilowatt-hour with the Diesel is from two to six times as high as with the turbine. This is due solely to the small output of the Diesel; an engine twice as large would have the labor cost reduced 50 per cent. The lubricating charges contradict the prevailing belief that the Diesel uses excessive amounts. That the oil engine is particularly adapted to central-station work is shown by the variation in efficiency with change in load. A 25 per cent change of load in the turbine produced a 20 per cent change in efficiency; on the other hand, the Diesel shows a 100 per cent variation with but 3 per cent change in the efficiency. It should be remarked that the average load on the steam turbine was 568 kw., while the engine's load averaged only 200 kw., or approximately half-load. If the Diesel had been fully loaded the labor and lubricating oil per kilowatt-hour would have been reduced by at least 6 mills.

Modern Steam Turbine Plant Costs.—It is somewhat different to obtain costs sheets of steam turbine plants with the various items segregated. In Table XVII is given an itemized report covering the costs in a 75,000 kw. steam turbine station. Coal was costing $7 per ton and this item can be corrected to suit local coal prices. It is obvious that a steam plant of even this capacity will find the Diesel a keen competitor.

Table XVII.—Production Costs in a 75,000 Kw. Steam Turbine Central Station

	Cost per Kw.-hr.
Superintendence, clerical and efficiency dept	0.08 mills
Boiler room operation, including ash disposal	0.20 mills
Boiler room repairs	0.15 mills
Engine room operation, including electrical	0.18 mills
Engine room repairs	0.12 mills
Miscellaneous expense	0.18 mills
Total cost operation, excluding fuel	0.91 mills
Cost fuel (coal at $7 per ton).	5.25 mills
Total cost per kw.-hr	6.16 mills
B.t.u. per kw.-hr. output	19,000
Pounds coal per kw.-hr. output	1.5
Kw.-hr. output per year	360,000,000
Tons coal used per year	270,000

This is considerably higher than the operating costs of the Diesel central station cited in Table XIII.

On the basis of $125 per kilowatt plant investment and a load factor of 66 per cent (which is abnormally high) the overhead charges per kilowatt-hour output of the steam plant would be 3.9 mills making the total costs 10.06 mills per kilowatt-hour.

Diesel Engines in a Cement Mill.—In 1916 the Lehigh Portland Cement Co. installed three 1,000-hp. McIntosh & Seymour Diesel engines in its Iola, Kansas plant. Each was direct-connected to a 750-kw. Westinghouse direct-current generator delivering 440 volts to the switchboard. In 1919 the fourth

Table XVIII.—Comparison of Efficiencies of Various Types of Power Plants

Heat Units in Fuel Consumed per Brake Horsepower per Hour

Load, per cent of rated hp.	Simple non-condensing Corliss engines		Compound condensing Corliss engines		Triple expansion steam engines		Steam turbines		Diesel engines		
	Engine rating, 100 i.hp. Boiler pressure, 100 lb. Steam per i.hp.-hr., 28 lb.	Engine rating, 400 i.hp. Boiler pressure, 450 lb. Steam per i.hp.-hr., 24 lb.	Engine rating, 200 i.hp. Boiler pressure, 125 lb. Vacuum, 25 in. Steam per i.hp.-hr., 15 lb.	Engine rating, 800 i.hp. Boiler pressure, 180 lb. Vacuum, 27 in. Steam per i.hp.-hr., 13 lb.	Engine rating, 800 i.hp. Boiler pressure, 150 lb. Vacuum, 27 in. Steam per i.hp.-hr., 12.5 lb.	Engine rating, 1,000 i.hp. Boiler pressure, 200 lb. Superheat, 100 deg. F. Vacuum, 27.5 in. Steam per i.hp.-hr., 10.5 lb.	Rating, 500 kw. Boiler pressure, 150 lb. Vacuum, 26 in. Steam per kw.-hr., 21 lb.	Rating, 5,000 kw. Boiler pressure, 200 lb. Superheat, 150 deg. F. Vacuum, 28 in. Steam per kw.-hr., 14 lb.	Engine rating, 150 b.hp.	Engine rating, 500 b.hp.	Engine rating, 2,200 b.hp.
100	52,500	41,500	25,500	21,000	20,000	17,500	21,000	15,000	9,000	8,400	8,000
75	60,000	47,500	29,000	23,500	22,500	20,000	23,500	17,500	9,500	8,900	8,500
50	79,000	59,500	36,500	29,000	28,000	24,500	27,000	20,500	10,800	9,800	9,000
25	138,000	99,000	58,000	45,000	43,000	36,000	36,000	28,000	15,400	13,000	12,000

For steam plants add allowance for stand-by, according to character of load.
To obtain equivalent pounds of coal divide B.t.u. by 12,500.
To obtain equivalent pounds of fuel oil divide B.t.u. by 18,800.
To obtain equivalent gallons of fuel oil divide B.t.u. by 143,000.

engine-generator unit of like capacity was installed, and a fifth engine was installed in 1920.

The cost per kilowatt-hour appears in Table XIX. This covers the operating charges, no fixed charge being tabulated. On a basis of an estimated investment of $600,000 and an overhead charge of 12 per cent, the power costs shown would be increased by 6 mills per kilowatt-hour.

The cement plant also has a gas engine which drives a portion of the mill and a small steam plant for the heating and fire-pump requirements. In Table XX are shown the total operating costs of the power plant for January, 1920. The heating and fire-pump costs are shown as totals.

TABLE XIX.—STATION COSTS

Total kw.-hr. output	1,190,100
Total fuel, gal.	112,203
Fuel cost per kw.-hr. at $0.032 per gal	$0.0030
Labor cost per kw.-hr	0.0008
Supplies and oil per kw.-hr	0.0015
Total	$0.0053
Fuel oil per kw.-hr., lb	0.697
Fuel oil per b.hp., lb.	0.182

TABLE XX.—POWER COSTS OF A CEMENT MILL

	Diesel plant	Gas engine	Steam power
Fuel cost	$3,608.44	$2,173.25	
Labor cost	1,806.30	622.32	$784.39
Stores cost	956.20	392.16	
Total cost	$6,370.94	$3,187.73	$784.39
Total power cost			$10,343.06
Total cement. bbl			1,096.86
Power cost per bbl			$0.095

Diesel Engines in Flour Mills.—Until 4 or 5 years ago the majority of the flour mills of the country were driven by Corliss engines. Although the power cost per barrel of flour ranged from 20 to 30 cents it was generally assumed that no other prime mover could compete with the Corliss engine. Increasing coal prices and higher labor charges now compel the mills to give more attention to power-plant economics. The favorable results secured in other industries where Diesels have been installed have caused a considerable demand for oil engines in the mills.

It is always difficult to secure actual data on Diesel plants and most of the information that has been published has been based on estimated costs. Fortunately the Page Milling Co. of Topeka, Kansas, have kept accurate data on the power costs of their Diesel-engined mill. Their figures are tabulated herewith. This mill contains a 385-hp. McIntosh & Seymour four-cylinder vertical Diesel which drives the mill machinery through a rope transmission system. The engine has been in operation for 4 years and runs 24 hours a day, being shut down on Sunday. During this period the load was varied from 50 to 80 per cent of the engine rating, the average load being about 65 per cent. Much of the excellent service secured is due to the fact that the engine has never been overloaded. In this matter the Page Co. emphasizes the necessity of always having the engine capacity greater than the load requirement.

For one year the power costs were as follows:

Barrels of flour, bbl .. .	124,000
Wages, per bbl. flour, in cents .	3 82
Fuel oil, per bbl. flour, in cents ($1.58 per bbl)	2 21
Lubricating oil, per bbl. flour, in cents	0 31
Incidentals, per bbl flour, in cents	0 11
Repairs, per bbl. of flour, in cents	1 73
Description 10 per cent per bbl. of flour in cents .	2.23
Interest and taxes, 8 per cent per bbl. flour, in cents	1 78
Total cost in cents	12 19

The item of repairs is somewhat larger than is customary. This covers the expenses of a man from the factory to install new camshaft gears and replacement of the steel support of the main driven rope sheeve.

The fuel oil was purchased at a contract price of $1.58 per barrel. This power costs per barrel of flour compares more than favorably with a steam plant which can hardly do better than 30 cents per barrel with coal prices at the present level. The repair charge will not exceed those of a steam plant. The wage charge is commendable as it indicates that the plant force are receiving wages that are equal to those paid steam plant engineers.

Moser Bros. of Great Bend, Kansas, have two 400-hp. Allis-Chalmers double twin Diesels in their flour mill. The cost of power per barrel of flour for one year's operation is shown in Table XXI.

TABLE XXI.—POWER COSTS FOR ONE YEAR

Fuel oil, gal. per bbl. flour	$0.55
Lubricating oil, gal. per bbl. flour	0.0033
Lubricating oil (bearings) gal. per bbl. flour	0.0016
Operating Costs	
Fuel oil per bbl. flour (6.3c. per gal.)	$0.0346
Lubricating oil per bbl. flour (oil at 60c.)	0.0036
Labor per bbl. flour (engineers and helpers)	0.0355
Repairs	0.012
Total operating cost	$0.0857
Overhead Charges	
Interest (7 per cent.) per bbl. flour	$0.0324
Depreciation (5 per cent.) per bbl. flour	0.0231
Insurance (2 per cent.) per bbl. flour	0.0093
Total power costs per bbl. flour	$0.1505

The capacity of the mill is 1,800 bbl. per 24 hours, or 540,000 bbl. per year. However, owing to the freight-car shortage during the period for which the power costs are shown, the mill was not able to operate to its rated capacity. The actual output was but 40 per cent of this amount. The power charges are, as a consequence, considerably higher than would have been the case with a constant output. The decrease in output affects all the charges with the exception of the fuel-oil and lubricating-oil costs. Operating under full load, this total of the power costs including overhead would be close to nine cents per barrel of flour. In calculating the fuel consumption, all kerosene and gasoline used in cleaning the engine and filters are included. The overhead charges are based on the total cost of the engine-room machinery, including the heating boiler.

Diesel Engines in Steel Works.—Due to the availability of large quantities of blast-furnace gas in the majority of steel and iron works this cannot be considered an attractive field for the Diesel. However, the Bethlehem Steel Co. has one Diesel plant and the Oklahoma Iron Works has a plant containing three Diesel engines. A view of its power plant is shown in Fig. 48 *A*.

Two-Stroke-Cycle vs. Four-Stroke-Cycle.—The majority of Diesel engines built up to the present time have been four-stroke-cycle units. The two-stroke-cycle engine was used by a number of builders early in the Diesel period but was abandoned by most of them. Recently attention has been turned again toward the two-stroke-cycle, and at present there are more large two-stroke-cycle units being built in the United States than four-stroke-cycle since for large powers the former possess certain advantages over the latter.

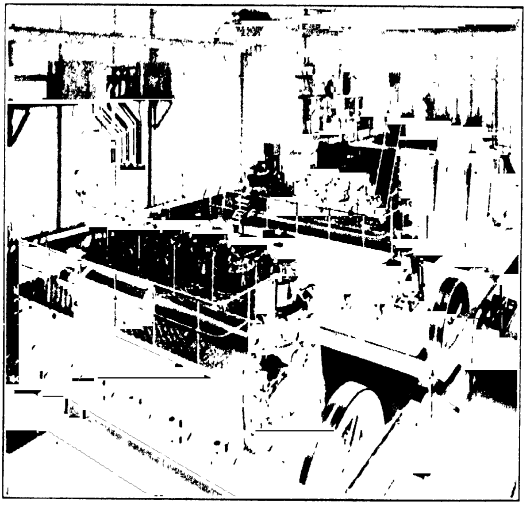


FIG. 48A.—Nordberg Diesels at Oklahoma Iron Works.

The four-stroke-cycle method requires the provision of an inlet valve and an exhaust valve for each cylinder, cam-opened and spring-closed, with a camshaft to carry the cams and gears to drive the camshaft from the crankshaft. It is obvious that, with a four-stroke-cycle method of charging, there is required a certain amount of mechanism which requires considerable care and attention, involving gears, cams, rollers, and rockers; these parts are multiplied by the number of cylinders, and these engines are usually multi-cylinder. These parts increase the amount of work required in manufacture.

In general, the market requires as inexpensive a unit as can be bought. The engine, however, must at the same time be well built and reliable, but cheapness in no case must outweigh reliability. In the larger engines, this demand immediately points to a two-stroke-cycle type engine where the weight per horsepower is low and the cost of manufacture correspondingly low.

As an offset to the lower manufacturing costs of the two-stroke-cycle engine the charging efficiency of the four-stroke-cycle unit is higher. By high efficiency is meant that the cylinder can be fully charged or more nearly fully charged with air, with the least expenditure of power in so doing, than the two-stroke-cycle method.

In addition the two-stroke-cycle engine requires a large air pump to furnish the scavenging and cylinder charging air. This pump calls for a large power input and reduces the mechanical efficiency of the unit.

The large amount of fuel consumed in the two-stroke-cycle cylinder, about twice that used in a four-stroke-cycle cylinder of equal bore and stroke, causes the heat stresses to be high. With correct cylinder head and liner designs this trouble has been surmounted.

It would appear that for large units the two-stroke-cycle, by reason of possible lower costs and in case of marine engines of less floor space, will likely be in greater demand. In a smaller type engine it is considered as yet questionable whether the two-stroke-cycle engine can be made more cheaply than a four-stroke-cycle. It will, however, not be a matter of price when the user of a small Diesel engine purchases it in preference to a steam plant. In any case, it must be a very reliable unit and for the same reason as expressed before; namely, that the class of labor employed to run such engines is inferior. There is no doubt but that for the same cost of building the two-stroke-cycle engine can be equipped with better material and better designed parts than can the four-stroke-cycle type. For this reason, it appears that the two-stroke-cycle principle will be adopted more generally in the smaller plants in the future than at present, always bearing in mind that it must be a better built engine.

It must be remembered that small cylinders present difficult charging and scavenging problems. It is generally conceded that the small two-stroke-cycle unit does not possess all the advan-sizes tages of the large units. Engineers are inclined to feel that in below say 500 hp. the four-stroke-cycle unit proves the superior.

CHAPTER VI

OIL ENGINE INSTALLATION

General.—The increasing number of oil engines being installed in industrial plants indicates that engineers will more and more encounter erection work involving the building of engine foundations. While this work merely calls for the same degree of care and skill that is required in the erection of any machine, there are a few points which if observed will make the work fairly easy.

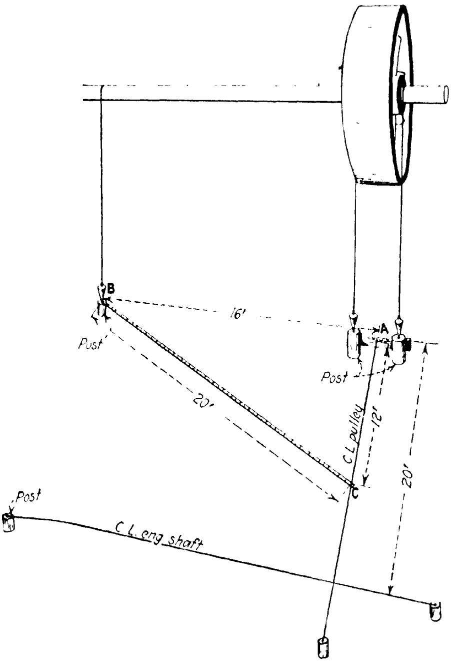


FIG. 49.—One method of locating engine shaft center line.

Establishing Center Line.—The first step is the locating of the desired engine and driving pulley center lines in relation to the shaft and pulley to be driven. Each installation has an individual problem as to permissible belt centers. As a general rule an engine from 50 to 100 hp. should be set at least 18 ft. from the driven shaft; on larger units 25 ft. should be the minimum. Two plumb bobs should be dropped from one side of the shaft, immediately against each side of the pulley to be driven. Posts should be set in below each bob and with the top a foot or so above the ground. Center tacks should then be put in marking the bob points. The shaft should next be revolved a half revolution to see if the pulley is square on the shaft. A board is nailed across the two posts mentioned and a tack, in alignment with the first tacks and at a point half the distance between two, is then put into the board. This gives a point that is on the direct pulley center line. The bob is next dropped over the shaft at a distance of 16 ft. from the center tack just mentioned; a tack located here now gives two points on the line of the driven shaft, as shown in Fig. 49, one of which is also on the pulley center line.

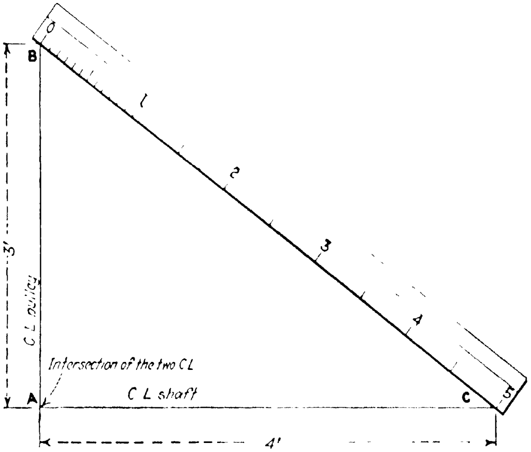


FIG. 50.—Locating lines at right angles by the 3-4-5 method.

A wire is now stretched from this point A at approximate right angles to the shaft center line. A mark C is made on this wire 12 ft. from the tack, as shown, and a steel tape stretched from the tack B to the mark C. The wire is shifted until the tape from B measures 20 ft. to the point C. This is the 3–4–5 method outlined in Fig. 50, wherein if 3 ft. be laid out on one line from *A* and 4 ft. on a second line measured from the intersection *A* then the two lines are at right angles if the diagonal *BC* measures 5 ft. Multiples of 3–4–5 may be taken, such as 9–12–15 or 12–16–20, as used in the aligning Fig. 49. The piano wire, on being set at right angles to the driven shaft, should be anchored to a post on the opposite side of the foundation site. The desired distance between the engine shaft and driven shaft is next measured, say 20 ft., and a stake or post set in with a center tack. The point thus determined is the intersection of the engine shaft and driving-pulley center lines.

The engine-shaft center line is next obtained by using the 3–4–5 rule. In making the measurements, the tape should be held as near horizontal as conditions will allow. Posts or stakes should be set in at some distance from the end of the foundation with center marks so that the line may be fixed for all time. An excellent plan, if the building permits, is to place four center tacks on the ceiling above the shaft and pulley center lines. This enables the engine to relocate the lines at any time. It is not necessary to attempt to establish an exactly horizontal center line, as this is best done after the engine is placed on the foundation.

Method on Large Engines.—With large engines the proper method is to use an engineer's transit in locating the engine's center line. The transit is set under the existing shaft and sighted along the shaft. Dropping two plumb lines from the shaft makes the locating of the line an easy problem. After the line of sight is established, and the datum mark made, the transit is turned 90 deg. and a stake set along the desired engine shaft line. Moving the transit to this latter stake, it is sighted on the datum stake or mark. This establishes a line of sight at right angles to the lineshaft. Turning the transit 90 deg. and driving a stake along the new line of sight gives two points in a line parallel to the datum shaft. After these two points are located, a strong piano wire run through them will indicate the desired engine-shaft center line. The anchorage for the wire ends should be substantial since workmen in moving material quite often strike it.

Foundation Form.—The foundation drawing showing the crankshaft and pulley-wheel center line is sufficient to enable one to lay out the foundation pit. If the engine is small, say under 50 hp., the dirt walls of the pit are quite suitable for the foundation form. If the dirt has a tendency to cave, rough planking may be placed in the pit, making a fairly good rectangular form. With larger engines it is advisable to construct a wooden form, for in most cases the foundation is not rectangular, but embodies an extension to carry the outboard bearing. In making a foundation, it is a good plan to provide a concrete trench or pipe chaise along one side of the foundation.

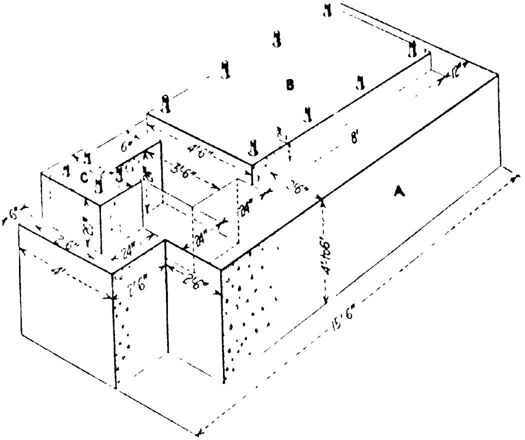


FIG. 51.—Typical engine foundation.

This allows all piping to be placed under the floor and yet be accessible. This is much better than the all too common plan of installing the pipe in the floor, permanently embedded in the concrete.

A typical foundation is shown in Fig. 51. At first glance the construction of the form may seem to be hard. Upon inspection it is found to consist of a rectangular lower portion upon which is set a portion which rises above the floor to carry the engine, and the block supporting the outboard bearing.

Frequently, a foundation plan embodies a great number of steps and angles, especially in cases where the draftsman is allowed to indulge in his fancies. Some claim that the earth pressure on the steps tends to eliminate vibration. However, the effect is slight and more is to be gained by employing generous dimensions and a better balancing of the engine. The engineer is advised to make the foundation as nearly rectangular as possible. Often this increases the concrete yardage, but concrete is cheaper than carpenter wages in building complicated forms.

The first step is to excavate the pit that is to carry the lower block, this being from 4 to 8 ft. deep, as the size of the engine and the character of the soil may determine. The easiest method is to make the pit rectangular, as example using the dimensions given in Fig. 50, 15 ft. 6 in. long by 7 ft. 6 in. wide, to which should be added 18 in. on each side to allow room for the forms and the removal of the planking. This gives over-all dimensions of 18 ft. 6 in. by 10 ft. 6 in. If there is no objection to leaving the form on the foundation after pouring, the allowance at the sides need be but the thickness of the plank plus that of the vertical supports.

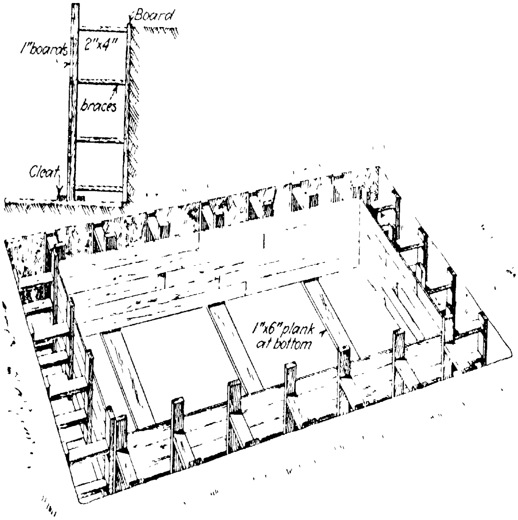


FIG. 52.—Foundation form in place

Proceeding with the form, 2 by 4 studding long enough to project above the floor level a distance equal to the height of the upper base should be placed against the pit walls at intervals of 30 in. and 1-in. boards, preferably tongued and grooved and finished on the foundation side, nailed to the 4-in. side of the studding. The boards should be cut to extend the entire length of each side of the pit and should be simply tacked with light nails just enough to hold them in place. With the sides made, a 1-in. plank should be laid across the bottom of the pit in line with each 2 by 4 and two cleats nailed at each end. When the sides are straightened, the base of each 2 by 4 rests between a set of cleats holding the bottom of the frame. Planking is next placed against the dirt sides in line with each 2 by 4, and short strips are joined to the plank and 2 by 4 timber. The ends of the form are next made in the same way. The foundation now has the appearance of Fig. 52, while at the side of the illustration is a sketch of the method of supporting the form. In addition a few braces should be nailed across the form at the top to brace the frame and to serve as a foundation for the upper base form. These braces must be no thicker than the amount of grouting to be used. The offsets in the lower form are next made by adding a box at each corner.

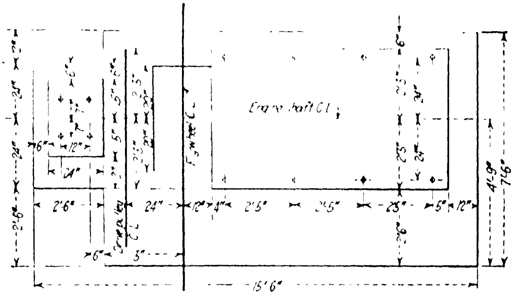


FIG. 53 Engine foundation plan

Engine Templet.—Before the upper foundation form is constructed, the templet for the foundation bolts should be made and placed over the excavation. It is usual for the engine builder to furnish this, but if the contract does not so specify, the engineer can easily make one up from the dimensions shown in the foundation plan. For a small unit 6-in. finishing lumber 1 in. thick is acceptable. For larger units there is not enough rigidity in such light material, and $1\frac{1}{2}$ by 8 lumber should be used.

Assume that the blueprints show the foundation bolts and engine shaft and flywheel center lines to be as shown in Fig. 53. Two planks about 12 ft. long should have the engine-frame bolt holes laid out and bored. Cross strips and a center strip at least 16 ft. long should then be laid on a smooth floor and a line drawn down its center. The longitudinal planks carry the bolt holes and are then nailed to the cross-planking, giving a templet as shown in Fig. 54. It is no great matter to place the planks carrying the engine-bed bolt holes parallel to the shaft center line. In locating the flywheel and pulley-wheel center lines, a steel square may be laid along the shaft center line and the flywheel line drawn at right angles. To check the accuracy, the 3-4-5 method may be used. The outboard bearing bolts are located in the same manner. The templet should be thoroughly reinforced and, after being placed on the foundation form, should be rechecked, for alignment of the center lines.

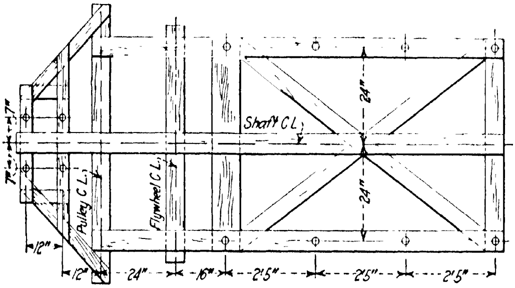


Fig. 54.—Making foundation template.

The templet is often swung from rods or heavy cable suspended from the roof. In this way, if the templet is shifted by accident, it will return to its true position. This method cannot be used in all cases, and the templet may be set on the form. A good plan is to drive in heavy posts some distance away from the foundation and suspend the templet from timbering nailed to these posts. This eliminates all damage of misalignment due to shifting of the foundation form. This method is largely followed but is not the best. The one method that is free from criticism is the use of two heavy timbers, about 12 by 12 in., to which are bolted cross stringers of 6 by 6 in. The templet is suspended from this framing by bolts, and, if the ends of the 12 by 12 timbers rest on solid footings the entire framing is rigid. If the engine is of medium size, 8 by 8 in. will serve for the bearing timbers. Such a structure will not move in event a barrow of concrete is thrown against it.

With the templet in place the foundation bolts should be hung from it and it should be lowered or raised until its bottom is on the level of the top of the upper form. The templet should now be fastened rigidly to the supporting posts previously mentioned. It is also necessary to level the templet. This is not difficult for by using a spirit level a close approximation of the true level is secured. A variation of ½ in. is not serious, as this can be corrected when leveling the engine bed.

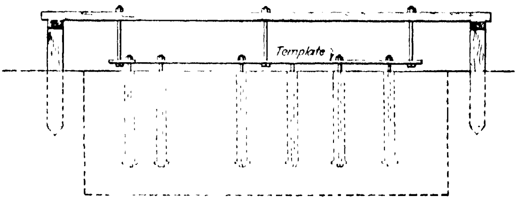


Fig. 55—Suspending template over foundation excavation.

Next in the process is the construction of the upper foundation form and the form for the outboard-bearing support. All that is necessary is to build up an open-bottom, open-top box at the proper place. The cross-timbers holding the top of the lower form are used as a basis of the framing for the upper form. No timbering should project into these open boxes, for if so, they will be covered with the concrete.

In the matter of the foundation bolts practice varies. It is necessary to have some play in the bolts, for the engine-bed bolt holes are cored and may be slightly out. A convenient tube to place around the bolts in order to have the bolts free to move is 2½- to 3-in. water spouting made of galvanized sheet steel. These are light and less costly than iron pipe. In large engines it is quite common to use such tubes together with recesses at the bottom, access to which is from the side of the foundation base. This has the advantage of allowing the engine to be moved onto the foundation without the necessity of being lifted to clear the protruding bolts. The bolts are then dropped in and a nut and washer placed at the bottom of the bolt as well as the top. The objection to this method is that it is hard to screw on the lower nuts since one must work in a very confined position. Preferably, the bolts should be set into the foundation before the latter is poured. Washers, say 6 to 19 in. square should be placed at the lower end of the bolt.

Foundation Material.—The proportioning of the concrete ingredients varies over a wide range. It depends to a considerable extent upon the character of the sand and gravel used. It is obvious that, if the gravel is not washed, the proportion of sand would be less than where a clean gravel was obtainable. For heavy foundations such as the one under discussion, a ratio of 1 part cement, 2 parts sand and 5 parts gravel, or crushed rock, is advisable. This mixture possesses ample binding strength and is free from the danger of cracking which develops when a leaner mixture is adopted. In order to lessen the foundation expense, many replace the gravel or crushed stone by broken brick. The objection to the brickbat lies in the danger of serious fractures; furthermore, unless they are clean, a good bond with the cement cannot be secured. Upon pouring the conglomerate into the excavation it should be thoroughly rammed, especially at the corners. Enough water should be added to make the mixture "quaky," but not enough to cause the cement to be lost by leakage.

In most installations the process of pouring the foundation extends over a period of several days. Each night the surface should be given a thorough wetting down to prevent any premature setting during the night. If the weather is cold, an old carpet or other covering can be placed over the foundation to prevent freezing. It is seldom necessary to place a foundation in freezing weather, but when such conditions do exist the water should be warm, and a liberal covering of straw and old carpets placed over it each night. After bringing the foundation to the desired level, the surface should be left in an unfinished condition and dampened each night. This keeps the surface concrete green and allows a good finish coating to be applied after the erection is complete.

Often the foundation print shows no reinforcing steel. With any oil engine there is need of tie bars inserted in the concrete. It is not necessary that special bars be purchased. Old steel rails, discard I-beams and the like are just as serviceable. A row of rods or bars laid longitudinally with crossbars at frequent intervals for a reinforcing matting will bind the entire structure. The steel should be laid about halfway from the base.

Vibration.—One of the objections voiced against the installation of a Diesel engine in an office building is the vibration so often present in the internal combustion engine. There is no adequate defense against this charge, for, as often installed, an oil engine sets up vibrations that can be felt even in large buildings.

In preparing the foundation for these installations a layer of felt at least 10 in. thick should be placed over the entire bottom of the foundation excavation. A concrete retaining wall 6 in. thick should be built about the foundation. This wall serves to keep the earth from touching the foundation. A wooden form for the foundation is then placed within this retaining wall. The form can be made of 2 by 4-in. studding and 1 by 12-in. rough boards. The 2 by 4's should not touch the retaining walls but should be supported by wedges. After the concrete is in the wedges can be removed; this will allow the wooden forms to be dismantled. The distance between foundation and retaining wall ought not to be less than 8 in. With this construction the foundation is not bound in any way to the building and the layer of felt will absorb all the shocks incident to the engine's operation.

Since the horizontal engines are, as a rule, more inclined to set up longitudinal vibrations due to the horizontal thrust, more massive foundations are required than for a vertical engine of the same number of cylinders.

With any concrete foundation, after the engine is erected a heavy coating of water proofing cement mixture makes an ideal finish. This coating will serve to keep any oil from seeping into the concrete.

Unloading Engine.—An engine is usually shipped on a flat car, although at times flat-bottom coal cars are called into service. If the latter, one end must be removed in order to get the engine off the car.

Various methods are followed by erecting engineers, the particular one used depending upon the plant location and other considerations. It is usually feasible to have the car set so that one end is accessible. A good plan of unloading is to construct a runway leading from one end of the car to the ground, as in Fig. 56, at an angle of not over 20 deg.; a steeper incline causse the engine to become unmanageable.

The engine frame, which in Fig. 56 is of the horizontal type, is lifted with ratchet or gear jacks and 6-in. maple rollers placed between the skids and car floor. Since the flooring is laid crosswise, 2 by 12 planking should be placed lengthwise of the car for the rollers to rest on. A $1\frac{1}{2}$-in. rope-tackle block should next be fastened to the frame at some convenient place, such as around the bearing housings, and the second block chained to the railroad track or drawbar. The rollers support the frame with ease, and men can roll the frame to the end of the car with little trouble after getting it started by using a couple of jacks tilted against the skids at the back end. When the engine frame projects over the car floor enough to tip, the tackle block, already mentioned, should be tightened so that the front of the frame will come down on a roller without jar. By easing off on the tackle the engine will roll down the incline without hitch, rollers being placed under the skids at intervals of about 3 ft., see Fig. 57. If maple rollers are not at hand, 3- or 4-in. pipe will do.

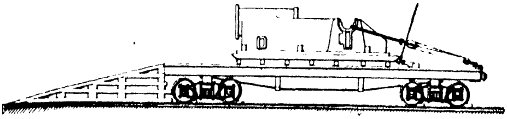


Fig. 56. Preparing to unload engine.

To move the engine onto the foundation, a runway of planking will make a good path for the rollers. Upon arrival at the foundation, the frame may be jacked up until the rollers can be moved onto the foundation or the runway inclined so that the rollers carry the engine onto the foundation.

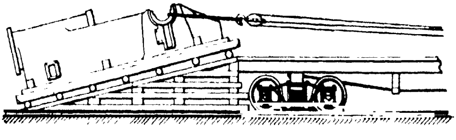


Fig. 57.—Unloading engine.

Lowering Engine on Foundation.—As soon as the frame is in the proper position, the skids are removed. By using jacks at three of four points the frame may be lowered onto a set of wedges. These wedges should not exceed 1 in. in thickness at the heavy end, and should be 3 in. wide by about 8 in. long. A jack will seldom lower closer than 2 in. of its footing, and so it is often necessary to make a depression in the concrete into which the jack is set. The hole need not be over 2 in. deep if step, or, as they are usually called, Barrett railroad, jacks are used. These will usually raise 10 tons without danger of breaking.

For heavier frames, geared or hydraulic jacks should be used. These will lower to about 2 in. of the bottom. The geared type of jack is to be preferred, since a man can lift 10 tons with ease. Figure 58 outlines the method of lowering the frame. Unless the frame weighs more than 10 tons, three jacks should be used, as this allows the leveling to be accomplished more easily than with four or more jacks.

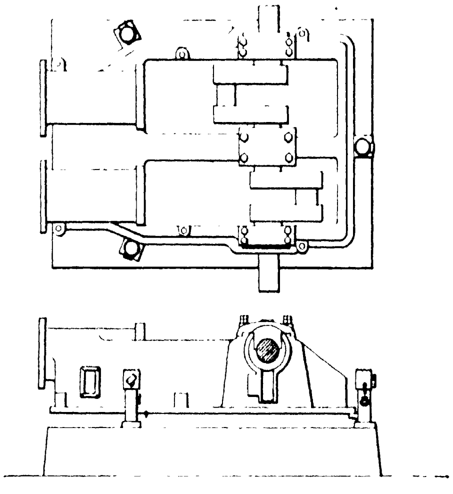


FIG. 58.—Lowering engine onto the foundation.

The same procedure is followed with a vertical engine frame. In many of the vertical engines, the frame rests on a base which is by no means difficult to handle. This bed or subbase must be placed on the foundation and leveled before the engine frame proper is set.

Leveling Engine Bed.—Many oil engines are provided with adjusting screws along the lower edge of the base, as in Fig. 58 *A*. In leveling up, iron plates about 4 by 4 in. are set under these screws. It is well to level the plates, grout them in and allow the cement to harden before the screws are brought down against the plates. If soda is mixed with the cement the grouting will harden in 10 min. By manipulation of the screws the frame is easily made level.

If the engine has a horizontal frame it is often leveled by placing a spirit level across the planed tops of the bearing housing and lengthwise on the housing. This will give a very close approximation of the true position. The erector must be careful that the engine-shaft center line lies along the center line established at the time of putting in the foundation when posts with center tacks were set in to establish both the shaft and pulley center lines.

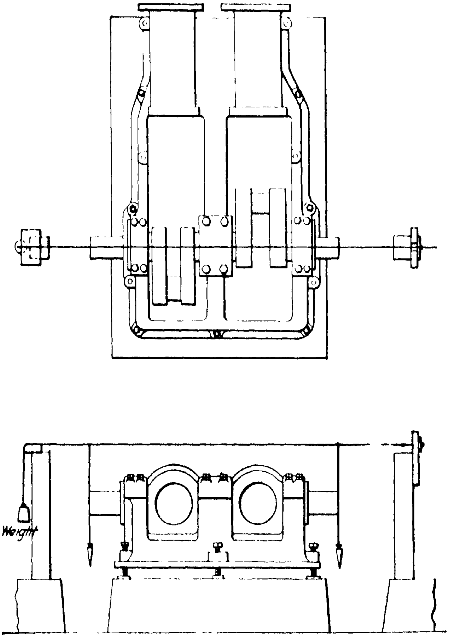


Fig. 58*A*.—Leveling engine with screws.

In all probability these posts are too low to allow a line to pass over the frame level with the top of the posts. Under these conditions center-line posts should be set in as close behind the first posts as possible without disturbing the latter. These posts should be high enough to permit the stretching of the wire above the engine, clearing the frame by, say, 20 in. The approximate location is easily determined. A wire should not be stretched between the two posts in a level position. This wire should be in the same vertical plane as the wire originally set up. The dropping of a plumb bob will make this easy.

The leveling of this wire can be done with a spirit level. A good method of attaching the two wire ends for leveling is to place a smooth piece of tin over the top of one post so that the wire can be shifted sideways or vertically. On the other post erect two substantial strips, as in Fig. 58 *A*. To this end of the wire is attached a circular piece of tin, while to the other end is fastened a 30-lb. weight, which is hung over the first post. The weight gives a uniform tension to the wire and brings the wire back to its position if anything displaces it. The tin washer on the second end, by being placed behind the two strips, may be raised and lowered, or shifted slightly sidewise. This will allow the wire to be placed in the exact vertical plane. In addition the tin washer may be raised or lowered to bring the wire level. The leveling of the center-line wire is done by placing the spirit level along the wire. One cannot hold the tool level without some support, such as a frame of planking or blocks. The level may be brought up by wedging, the wire's position checked, and the tin disk tacked into place.

Establishing Center Line. - After this wire is put up, two wires should be laid at right angles to the lineshaft that is to be driven by the engine by the well-known 3-4-5 rule, and the distance from the driven shaft to the shaft center-line wire measured for the purpose of checking the position. A steel tape should be used, as a linen tape is unsafe, stretching as much as 1 in. in 25 ft. The pulley center-line wire should be stretched if it has not already been established. Often the position of the pulley is marked on the shaft; if not, the position is indicated by the keyway.

Dropping a bob to the pulley center line will show how much the engine must be shifted sidewise to come into the correct position. To bring the engine shaft exactly under the shaft center-line wire, erected as noted above, putty should be hammered into the shaft-line centers and a small mark made in the exact centers. This is found by calipering the shaft. If plumb bobs are dropped from the center wire, the erector will be able to shift the engine the slight amount needed to bring these centers square with the bobs. Since the weight will always bring the wire back to its level position, the bobs may for convenience be hung from the wire. A good erecter, with the foundation bolts to guide him, should have the engine in a position that will require but little shifting. The next step is the leveling of the frame. Since the shaft is bedded in the bearings in its proper place, all that is necessary is to level the two ends of the shaft and then level the frame lengthwise. By measuring from the wire to the shaft with a pin gage the latter can be brought level with the wire. If the engine is of any size it is well to remember that the long wire will not be exactly level, having a small sag in it. Figure 59 is a chart showing the allowance that should be made for a 0.16 wire with different lengths between posts.

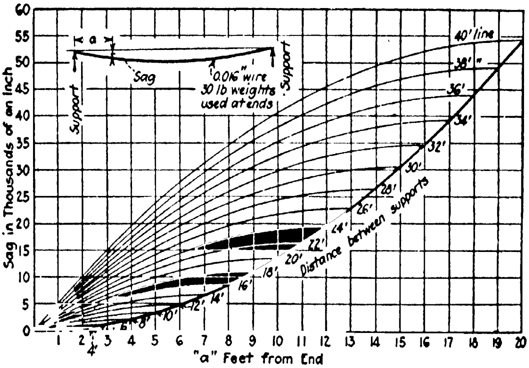


FIG. 59 —Deflection of steel wire.

If an engineer's level is at hand it may be set up at a point somewhere back of the engine. The instrument is then sighted at a rod placed vertically on one end of the engine shaft. A mark, cut by the hair line of the engineer's level, is made on the rod. Next the engineer's level is swung so that it will be in line with the mark when the rod is held on the opposite end of the shaft. One end of the frame must be raised or lowered until the mark again cuts the cross-hair. The leveling should be rechecked at least once. It is seldom, though, that such an instrument is available, although for engines over 500 hp. it should be used.

To align the frame lengthwise of the cylinder bore, it should be sufficient to level the bearing housing, although a safer procedure is to run a center line through the cylinder.

Leveling Small Engines.—A customary method, which, while apparently crude, is often followed successfully on small engines, is the leveling by means of a spirit level with a groove in one surface. This is placed on the engine shaft, and by the cross bubble brought up in the proper position. Longitudinal alignment of the shaft is then indicated by the second bubble.

Leveling Vertical Engines.—The leveling of vertical engines calls first for the placing of the subbase on the foundation, and its leveling. Engines as a rule have gage marks on the subbase immediately below the shaft center line. With these gages no difficulty is experienced in leveling and setting the base in its place. In the absence of such gages, a board may be placed in the two outer bearings with a center line in each board. Still another method is to locate the center of the bearing and make a scratch on the face of the bearing at the bottom. This gives the center plane, and measurements can be taken from the bearing bottom rather than the shaft center. After the engine base or subbase is properly leveled and set, the foundation bolts

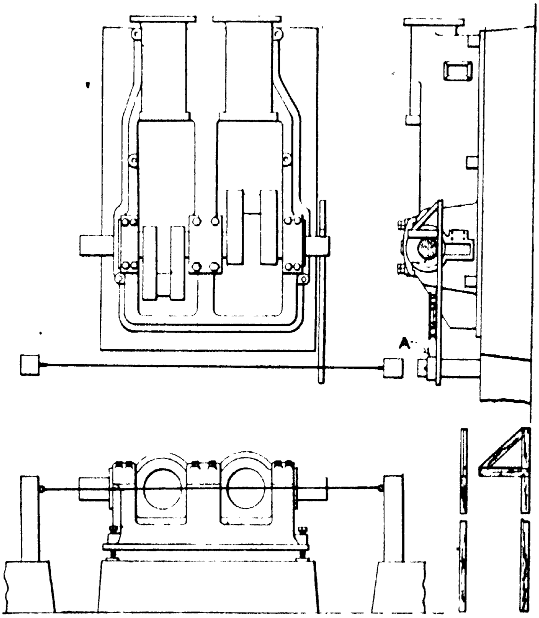


FIG. 60.—Easy way to align engine.

should be drawn up against the frame, but no great pressure should be used, for there is danger of distorting the frame.

Vertical engines of 200 hp. and less, especially high-speed units, are almost always shipped with cylinders in place and the shaft in the bearings. It is costly to remove the cylinders for aligning purposes. An excellent method of leveling this type of engine, and one often used in horizontal units, is the running of a center line along one side of the foundation, parallel to the driven shaft. All that is required is to set the engine so that both ends of the shaft are equal distances from this line and that the shaft bottom at each end is the same distance below. A leveling bar as shown in Fig. 60 is made, preferably of hard lumber, being glued and screwed at the joints. This is placed under the shaft and in contact with it at the side and bottom, and is provided with the block *A* equal to one-half the shaft diameter. A spirit level set on the bar will show when the shaft is level with the wire, while a mark on the bar will allow the engine to be shifted until both ends of the shaft are equally distant from the wire. This is an excellent method, and has the advantage of removing the center line from the engine. If the shaft is not of the same diameter at the ends, accurate spacing blocks placed on the bar to equalize the discrepancy will do nicely. For engines of less than 300 hp., this leveling method is the most practical of all. It is not necessary to have the wire level with the desired shaft center line. All that is required is that the line be level and parallel to the future engine center line.

Grouting the Engine.—Before running the grouting, boards should be placed around the foundation, extending above the surface the desired depth of the grouting. A mixture of one part sand and one part cement should be thoroughly mixed with enough water to make it flow readily. This is poured in around the base and troweled into the center of the foundation. The space between the bolt and iron piping should also be filled. Each bolt-hole should be filled completely, and the ordinary laborer should be watched to prevent him from covering the top of the foundation before fully filling the holes.

Enough grouting should be poured in to fill the space up to the top of the boards already mentioned. As the cement sets, the boards may be removed and the grouting troweled down at a bevel. After this has set for several days, the foundation-bolt nuts should be drawn up tight and the iron wedges knocked loose, or, in case of screws, these should be backed off a little.

With vertical units the engine base or A-frames must be placed on the subbase after the latter is leveled. This is not difficult, and it is merely necessary to set the base or A-frames so that the drowel pins can be driven into their recesses.

Putting on the Cylinders.—Continuing on the erection, the cylinders are placed on the base. If an I-beam trolley and a suitable chain hoist are in the plant the handling of the cylinders is easy. Since the cylinders must be centered over the studs on the top of the base they must be held exactly vertical by the hoist to prevent jambing on the studs. The author recommends that a 6 by 6 timber be cut long enough to extend over the cylinder top. Holes large enough to accommodate the cylinder head studs are bored in this timber, and the latter slipped over two opposite studs, and the nuts screwed on. By wrapping a chain around the center of the timber and engaging the chain in the hoist hook, the cylinder can be lowered over the studs without trouble. This is shown in Fig. 61. Every plant should be equipped with an I-beam trolley and differential hoist. This hoist should be of at least 5-ton capacity. The I-beams can rest on brick pediments that are incorporated in the building walls. In case such equipment is not available, it becomes necessary to construct a wooden frame similar to Fig. 61 *A*. The top beam should clear the frame by at least 10 ft. to give ample room for the blocks. If timbers of sufficient length are not obtainable the frame can be made of built-up timbers constructed of six planks 2 by 2 in. in size, lag-screwed together. Stout planking should be placed along the side of the engine to prevent the

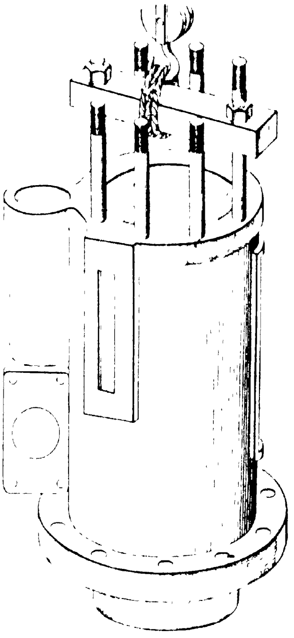


FIG. 61 Method of handling cylinders.

cylinder damaging the frame as it is hoisted to the top; 6 by 6 cross timbers can be laid across the top of the frame to act as a landing for the cylinder. These timbers guard the studs against damage. The tackle blocks, or chain hoists, which should have at least 5-ton lifting capacity, are fastened to the cylinder top, and it is raised until it can be lowered onto the 6 by 6 in. platform. It is always advisable to use two blocks. This is a safety measure in case one breaks. After the cylinder is placed on the platform it should be pinched into place over the studs. All the timbers, save two outside the bolt circle, can be removed. By lifting the cylinder with the blocks, which have been centered above the cylinder, the two timbers are withdrawn and the cylinder lowered onto the frame.

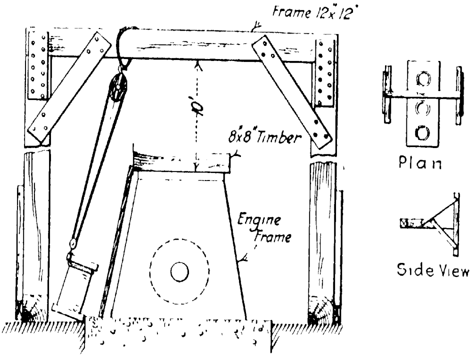


FIG. 61A.—Using A-frame in absence of chain hoist.

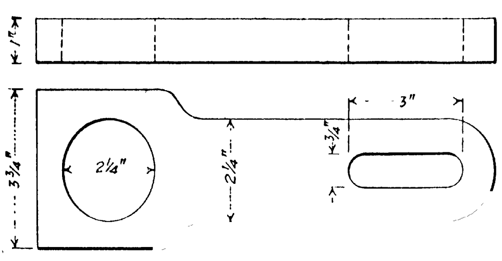


FIG. 62.—Centering strip.

After the cylinder is in place a plumb bob is dropped through it. To do this a metal or wood strip, similar to Fig. 62, is fastened to one of the cylinder-head studs. A washer is attached to the bob line after the line has been passed through the slot in the strap. The strap is placed over the approximate cylinder center. The slot allows the bob line to be moved until it is exactly center. To center the plumb line at the top of the cylinder, a pair of inside calipers are used. This centering calls for the greatest patience on the part of the erector. After centering the line at the top, measurements must be taken at the bottom of the cylinder to ascertain if the bob line is central at this place. Since both the cylinder flange and frame top are machined true, it is seldom that the cylinder does prove out of plumb, providing the frame has been properly set. If this should occur, the frame must be wedged up until the cylinder center line is plumb. If the engine is multi-cylindered a plumb is dropped through each cylinder and kept in place until the engine shaft is lined up, Fig. 62*A*. The bob lines must square with the shaft center line. Practically the same method can be followed with the A-frame engine. Many engineers do not check the cylinder alignment, feeling that the factory has already done this and has properly doweled the castings in place, but it pays in the long run to check it.

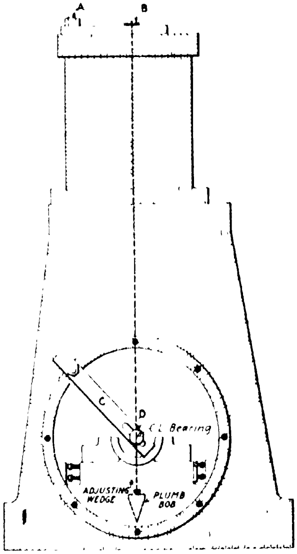


Fig. 62*A*.—Aligning cylinder and shaft.

Centering the Shaft.—It is necessary to check the crankshaft bearings to insure that the shaft will be square with the cylinder center lines. Two metal or wood strips, should be fastened to the ends of the engine frame, as in Fig. 62 *A*, with a piano wire stretched between. The disks should be moved until the horizontal wire just touches the three cylinder plumb lines; the wire should be leveled at the same time with the spirit level. The bearings, which have been placed in their housings, should be calipered to see if the line is central with each bearing. If a bearing is out, then it must be shifted until it registers central. With the wedge-type bearing this alteration can be obtained by proper movement of the wedges. If the bearings are of the non-adjustable shell type it becomes necessary to scrape the high bearing and shim up the low ones. In most engines the bearing housings are bored in alignment, and it is only necessary to see that the bearing shells are of the same thickness. This makes each bearing in alignment, and the cylinder lines must check if the cylinders have been properly fitted.

Bedding the Shaft.—Before the shaft is lowered onto the bottom bearing shells small lead wires about $\frac{1}{32}$ in. in diameter should be placed along the bottom of each bearing. When the shaft is placed in position these wires flatten out. By raising the shaft these wires can be removed and their thickness measured by a micrometer. If the lead is of the same thickness throughout its length, the shaft evidently bears evenly in a longitudinal direction. If any unevenness is present the shell must be scraped to a fit. To check the area of the shell which is supporting the shaft, the shaft can be coated with Prussian blue and rotated. Upon lifting the shaft the bearing will show which points are high or low. Scraping the babbitt will bring the surface to a perfect contact.

Many erectors have the habit of using a chain sling on the shaft. This is absolutely out of place with a Diesel engine. Sacking wrapped about the shaft and the use of rope slings will eliminate all danger of scoring the shaft. It requires only a small dent or cut on the journal to ruin a bearing. In handling the shaft it is advisable to suspend it by slings from each end; if possible, the slings should be tied on the shaft at other than journal joints. If no traveling crane is at hand, two wooden forms, such as indicated in Fig. 61 *A*, can be constructed.

Aligning Outboard Bearing.—With an engine employing an outboard bearing the alignment of the extension shaft merely necessitates the continuation of the shaft center line. In erection the main point that demands the exercise of extreme care is the bolting of the extension shaft to the main shaft, or to the flywheel, as the design may be. The flange bolts should be drawn up uniformly, a part turn of each nut in succession. Some erectors are in the habit of bolting the two parts together with the outer end unsupported by the outboard bearing. They then turn the engine over and adjust the outer bearing against the shaft end. The danger here lies in the liability of the unsupported shaft weight throwing it out of level. Many hot outer bearings can be attributed to this carelessness.

In leveling the extension shaft and outboard bearing, erectors follow various methods. A good plan is to raise the outboard bearing until the shaft no longer touches the main bearing next to the flywheel. This journal is coated with Prussian blue and the outer bearing is then lowered until the blue is rubbed off by the lower shell when the shaft is rotated a half turn or so.

Putting on the Flywheel.—The work of putting the flywheel in position is often difficult, due to the building arrangement. On the basis first that the unit is a belted machine, the extended shaft is set in place and the out bearing aligned, after which the shaft is removed. One-half of the flywheel should next be skidded to the side of the foundation and in front of the flywheel pit. The half is then up-ended, and by attaching rope slings to the arms the casting is lifted and swung over the pit. Cross blocking is placed in the pit and the wheel half is lowered until the hub is approximately in position.

The shaft extension is next swung into place. If room is available it may be shifted lengthwise into its position. Unfortunately the usual plant has little room at the end of the engine and the shaft must be rolled along the engine until it is opposite the flywheel. It may then be lifted enough to clear the flywheel resting in the pit and after cross timbers are placed under the shaft the latter may be rolled into position. With the shaft supported by the hoist the cribbing is removed until the shaft half of the coupling fits into the engine coupling half, and the shaft end rests in its bearing. Before tightening the coupling both the alignments of the shaft should be checked. Some engineers merely shift the bearing until the extended shaft shows no sign of wobbling but this is by no means good practice.

Installing the Generator.—With a unit direct-connected to a generator the procedure follows the same general lines. The shaft, however, must be placed high enough above its true position to enable the generator frame to be moved onto the base or rails. This requires the blocking of the shaft and rotor in such a way that the generator can be shifted over it. As soon as the shaft end is exposed blocking should be placed under it. When the generator is moved up to the position of the rotor the shaft should be shifted until the generator will pass over the rotor, without cramping. To do this, blocking should be placed in the generator pit until the generator can be moved over the pit. Of course, if the generator sets level with the floor or above it, there is no difficulty, for the shaft may be placed in exact position before the generator is moved. When the generator sets in a pit the lowering of the frame, shaft and rotor at the same time is somewhat difficult. There is enough air gap between the rotor and frame to allow a little leeway.

The best plan is to run down the leveling screws on the generator feet until they support the generator. Part of the blocking under the generator is then removed. The shaft supported by four screw jackets placed on the cribbing, each pair of jacks, supporting a block upon which the shaft rests, being kept from rolling by cleats. The whole may now be lowered until the generator rests on its sole plates, the shaft coming down into place. The shaft must be reblocked to allow the end jacks to be moved and the outboard bearing slipped into place. The shaft coupling should be connected and the alignment of the shaft and outboard bearing checked.

The next step after bolting up the coupling, fastening the outbearing and centering the generator is to put on the top half of the fly wheel.

The lower half is lifted into contact with the shaft at the hub ànd blocked into place. The top half is moved opposite the wheel pit, upended and a slide placed under it. The casting is next lifted by jacks and a cribbing built under it. It is then moved into place over the shaft and lowered until the hub rests upon the shaft in line with the bottom half of the wheel. The hub bolts should now be drawn up as tight as possible.

Putting in Wheel Links.—The rim links should next be heated in a forge or like place. When at a dull red heat the links are slipped into the recesses in the rim and driven home. As soon as they cool the rim halves are drawn together. It will be found that the hub bolts are now slack. It is often impossible to draw them tight. The best scheme is to remove the bolts, and after warming the bolts and nuts the nut may be drawn up sufficient to make the bolts tight when the latter shrink upon cooling. Both bolts should be drawn up at the same time to prevent tightening the hub on one side only. When the wheel is in place the supporting cribbing is unwedged and removed.

Installing Parts.—After the engine frame and generator are in place the pistons and connecting rods are bolted up. If the engine is direct from the factory the correct amount of shims for running conditions are already between the bearing halves. Likewise the babbitt shells have been scraped and bedded in. Old engines or engines being overhauled must have the bearings examined and fitted to their respective pins or journals. After some years of service bearings, especially the main shaft bearings, take on a hard skin or surface, which is removed with difficulty. This is more often found in engines where a splash system of lubrication is used. The fine carbon apparently gets into the bearings, and settles, cutting the journals in some instances. The proper running clearances, etc., are taken up in the chapter on Bearings.

The installation of the several valves and cages is comparatively easy. The chief point is to see that the ground or gasketed joints are in good condition and that the cages are not distorted by sledging up the holding-down bolts.

Air Lines.—The connections of the air lines to and from the air receivers deserve a great deal of attention. The pipe is special, equal to or better than double extra-heavy, and in threading perfect threads must be made. As few joints as possible should be placed in the lines, and provisions should be made for drainage before the lines reach the receivers or bottles, although seldom is this done. If any dope is used, a mixture of litharge and glycerine is the only preparation with which the threads should be coated. It is excellent practice to tin the threads, and after heating the fitting, screw the parts together while the fitting is still hot. Many fittings are recessed for a babbitt or lead filler—this is also a good scheme.

Oil Lines.—The oil lines should be put together with good clean threads. No white or red lead should ever be used, for the oil will soon saturate the lead and leak. If heavy fuel oil is to be burned it must be warmed by some means in order to raise its fluidity to a point where the oil will pump freely. Likewise a small overhead oil storage tank should always be installed. This insures a supply for starting, and also gives a slight pressure head to the fuel on its way to the pump. All danger of gas or air binding of the engine fuel pump is avoided.

When a heavy oil is to be used it is a good idea to have a small tank of kerosene for starting purposes. In this way the engine will start promptly, and after a few minutes may be shifted to the heavy oil. The main fuel-oil supply should be stored outside of the building, at least 50 ft. away, to avoid fire hazard. To handle the oil from this tank to the small overhead tank a motor or hand-driven rotary pump is essential. If the unit is over 200 hp. the pump should be motor-driven. Details will be given in the chapter on Fuel Oil.

One of the most important accessories in the plant is a suitable lubricating oil system, including a purifying and reclaiming outfit. This should never be omitted on the grounds of first cost, for it will return its total cost in two years or less in the oil saving alone. Either a centrifugal separator or coagulating filter is good.

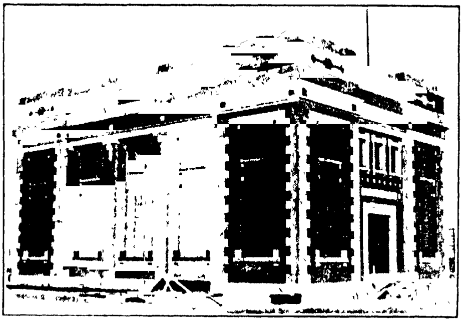


FIG. 63.—Exterior of a 3,000 k w. Diesel plant

Tools.—Many other features will be touched upon in the several following chapters. Before closing this discussion mention must be made of tools. A plant worthy of its name should be fully equipped with a good supply of drill bits, hammers, chisels, punches, scrapers, blow torches, soldering outfit, drill press, lathe, electric drill and electric grinder, files, screw-drivers, levels, calipers up to the piston diameter, and such tools as the engineer will find necessary. A board carrying all tha wrenches furnished with the engine should be placed on the well close to the machine. The small tools should be kept in a set of drawers or glass-front shelves in the engine room or tool room. The successful engineer in either a stationary or marine

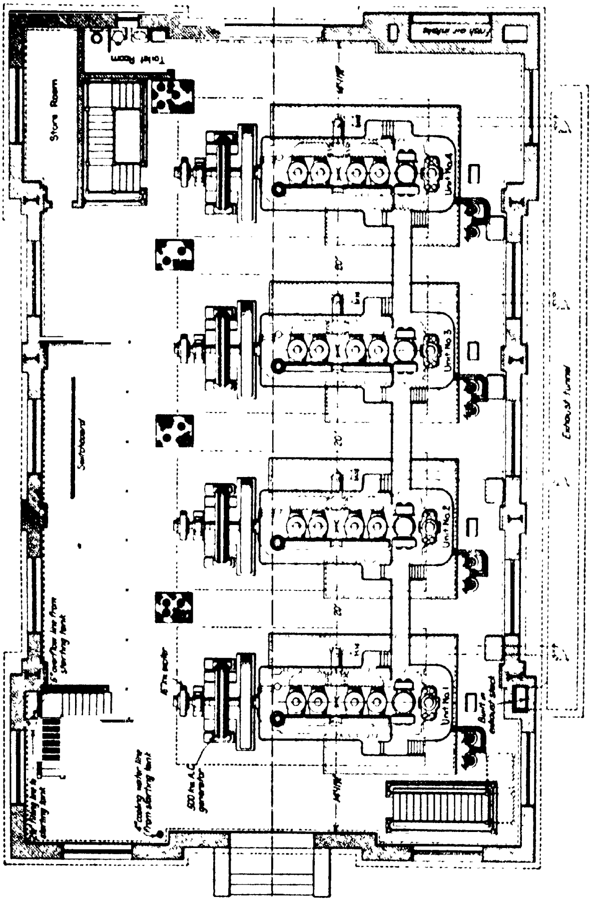


FIG. 64.—Plan of a 3,000 k.w. plant.

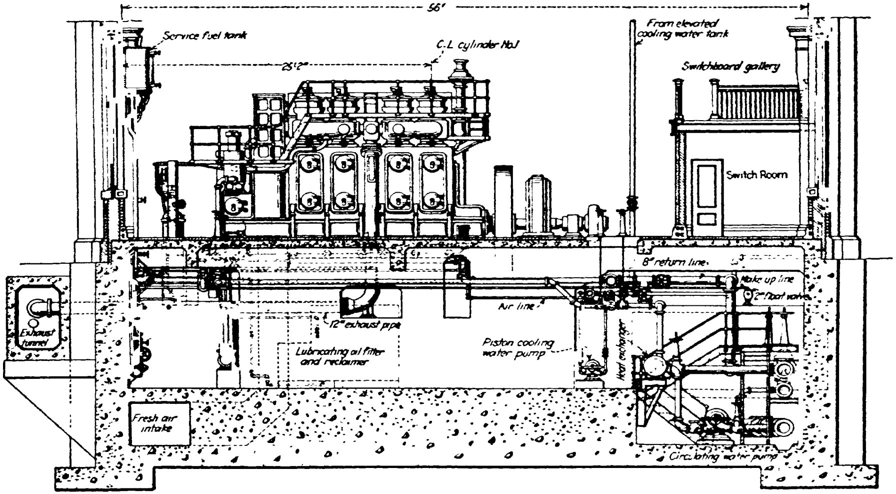


FIG. 65.—Cross-section of plant shown in Fig. 64.

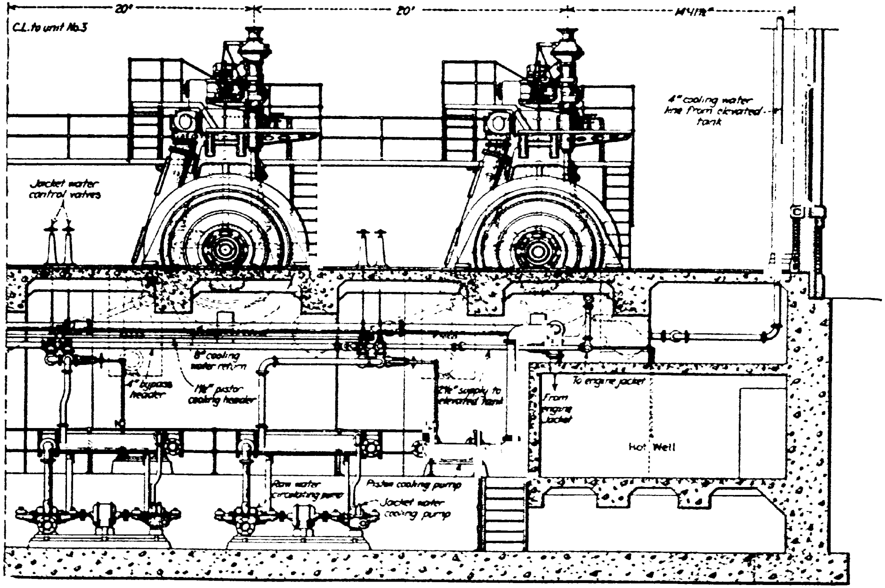

20'
20'
14'4½"
C.L. to unit No. 3
Jacket water control valves
4" cooling water line from elevated tank
8" cooling water return
4" bypass header
1½" piston cooling header
2½" supply to elevated tank
To engine jacket
From engine jacket
Hot Well
Raw water circulating pump
Piston cooling pump
Jacket water cooling pump

FIG. 65A.—Cross-section of plant shown in Fig. 64, showing two 750 hp. Busch-Sulzer Diesels.

plant should have a place for everything, and a set time to do all the routine work.

Plant Building.—The character of the plant building is largely determined by the machine investment and the locality. Obviously the type of building suitable for a 100-hp. engine would

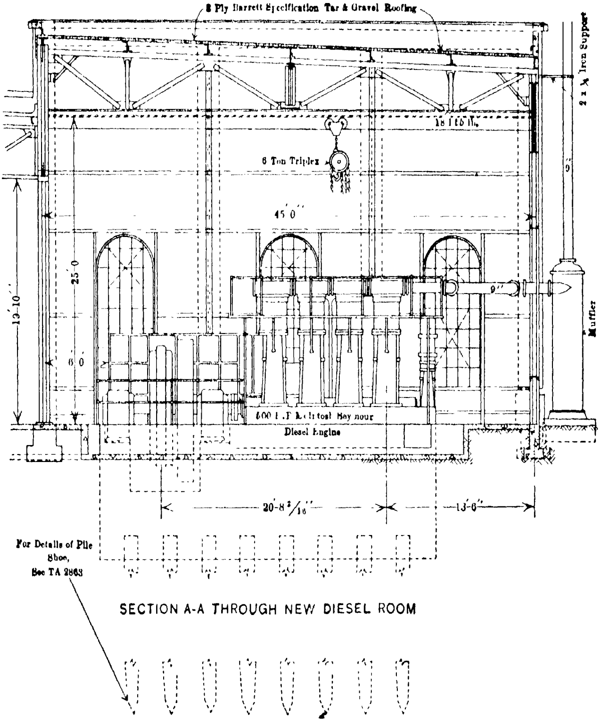

8 Ply Barrett Specification Tar & Gravel Roofing
2 x ... Iron Support
18 I 65 lb.
6 Ton Triplex
45' 0''
25' 0
13'-10''
6' 0
9''
Muffler
600 H.P. McIntosh Seymour
Diesel Engine
20'-8 3/16''
13'-0''
For Details of Pile Shoe, See TA 2863
SECTION A-A THROUGH NEW DIESEL ROOM

Fig. 66.—Piling for engine foundations.

not be fitting for a 3,000-hp. plant. In Fig. 63 is shown the exterior of a Diesel plant of the Chicago Sanitary District, while Figs. 64 to 65 *A* show the plan of the machine layout, there being four 750-hp. engine-generator sets. Figure 66 outlines a plant extension for a 500-hp. Diesel, together with the method of piling the foundation.

CHAPTER VII

DIESEL ENGINE FRAMES

Horizontal Diesel Frames.—The horizontal Diesels built in the United States are of the center-crank box design. While gas engines have been built with overhung cranks, the Diesel builders have preferred to support the crankshaft with two main bearings and an outboard bearing. It is generally understood that this construction makes a safer support. If, however, the three bearings are not kept in alignment, the bending stresses may be greater than in case of a two-bearing support.

There is little that differentiates the several makes of frames; the weight per horsepower is fairly uniform; and the degree of depth, width and length of frame of the several engines closely agree.

For a time after the general introduction of the Diesel engine in the United States, operating engineers were more favorable to the horizontal frame. It was believed that the engine was more accessible than the vertical design. There is no disputing the fact that pistons may be removed with less trouble from the crank end of the cylinder. Likewise the valve-gearing, etc. are within reach but the objectionable feature of poor balancing has hindered the growth of the horizontal engine. In addition, when going into large powered units, the vertical engine, due to a better arrangement of cylinders upon a single bed plate, will take up less room fore and aft.

At present the drift toward the vertical engine is almost universal. Only three firms are actively building the horizontal engine, although there are several different makes in service. The horizontal Diesels in service include the Allis-Chalmers (Fig. 20), De La Vergne type FD (Fig. 23), National Transit (Fig. 21), McEwen (Fig. 22), Snow (Fig. 19), and Standard (Fig. 32).

The Standard Diesel Frame.—Being a two-stroke-cycle engine with a stepped piston, Fig. 77, this engine has its frame enlarged at the cylinder end to accommodate the enlarged portion of the piston. This engine, however, has been replaced by the Standard's vertical design.

Vertical Diesel Frames.—The vertical Diesels fall into Box frames, A-frames, Staybolt, and Truss frames. The A-frame was the original design adopted in Europe, and the first Diesel built in the United States (see Frontispiece) was of this type. This was adopted by two or more builders although at present only one manufacturer adheres to this frame.

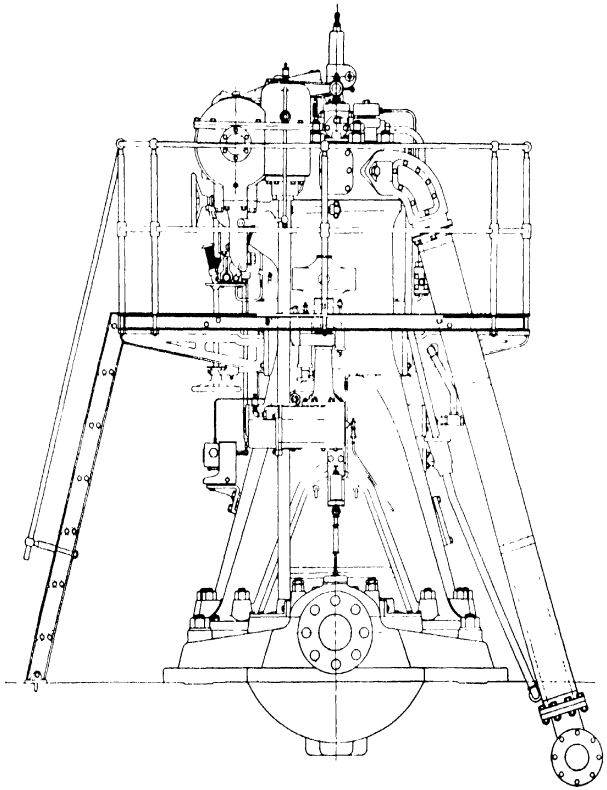


FIG. 67.—Frame of Fulton Diesel.

Fulton Iron Works Engine Frame.—The Fulton Iron Works, when it took up the Tosi Diesel, make use of the Tosi A-frame;

the Fulton frame appears in Fig. 67. As the illustration shows, the frame is extremely heavy, weighing more than that of any other Diesel.

McIntosh & Seymour Engine Frame.—The Diesels built for several years by the McIntosh & Seymour Corporation employed the A-frame design, as will be seen upon examination of Fig. 83. Of recent years, since 1918, the box frame, Fig. 68, for stationary engines up to 1,000 hp. has been used although at present 750-hp. Diesels with the McIntosh & Seymour Marine Diesel framing have been installed in stationary plants. The McIntosh & Seymour A-frame was the first introduced in other than experimental Diesels in the United States, at that time all others were either box-frame or vertical engines.

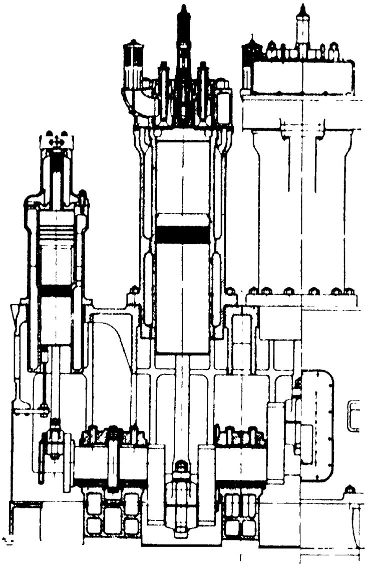


FIG. 68 — McIntosh & Seymour box frame.

While it is impossible to enumerate all the reasons for the drift to the box type of framing, undoubtedly the two important factors have been the item or replacement and the question of accessibility. In case of cylinder jacket fracture, which has occurred at times, the entire A-frame must be replaced or welded, while with the box-frame only the cylinder casting need be repaired. Besides, the A-frame makes the handling of the crankpin bearing most difficult, for the space around the crank is by no means liberal.

Nordberg Type VE & EG Diesels.—The type VE and EG two-cycle Nordberg engines have A-frames wherein the cylinder jackets are cast with the supporting legs or A-frame. The VE engine is shown in Fig. 68 *A*. Cross heads are used on the engines, the guides for which are cast as pads on the A-frames in preference to the customary plan of having bolted guide plates.

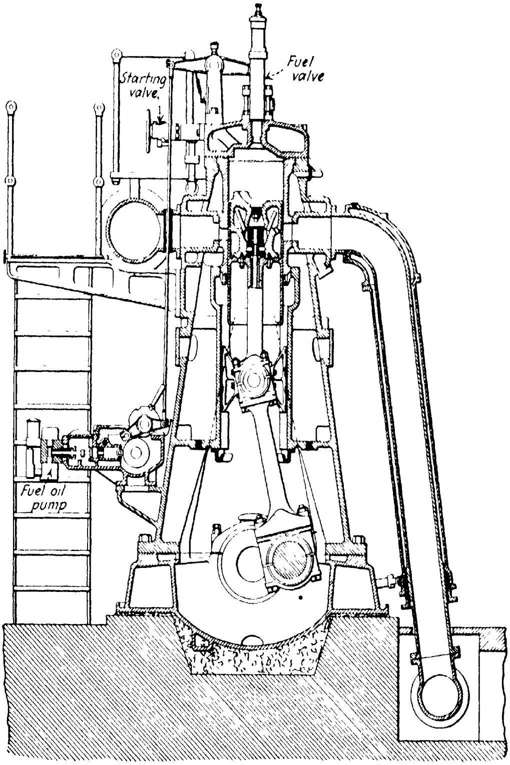


Fig. 68*A*.—Nordberg Type *V.E.* frame.

Dow Diesel.—The Dow Diesel design, Fig. 69 embodies the A-frame. The cylinder liner is pressed into this casting which in turn is mounted upon a heavy subbase. The shaft bearings are seated on the cross members of the subbase.

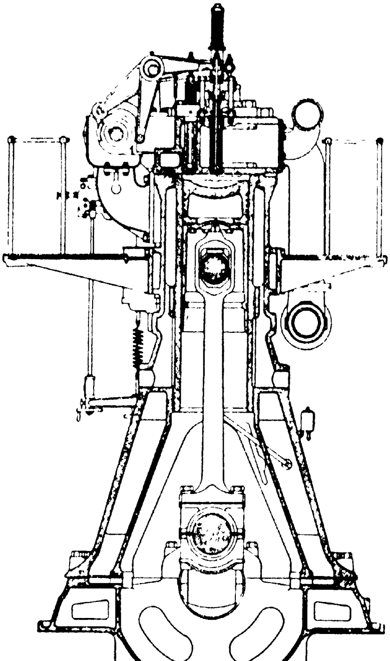


Fig. 69.—Frame of Dow engine

McIntosh & Seymour Box Frame.—The box frame engine appears in Figs. 17 and 68. The frame consists of a low subbase upon which rests the frame proper, which is in the form of a box. Immediately in front of and behind each crank is a large inspection door. The shaft bearings are supported on cross girders which are cast with the lower or subbase. Small openings in the main frame in line with the bearings permit the inspection of the bearings while the engine is in operation.

Busch-Sulzer Type A.—This engine is no longer manufactured as their present engines are of more modern designs. There are, however, practically as many of the Type A and their forerunner, the American Diesel, in service as any of the later-day units. The frame is a heavy box with tension rods reaching from the bottom to the top, thereby taking the working stresses off the cast-iron box, see Fig. 13.

Busch-Sulzer Type B.—The Busch-Sulzer manufactures the type B, four-cycle Diesel in sizes from 120 to 500 hp.; large sizes being two-cycle engines.

The type B-frame is shown in Fig. 69A, and upon examination it will be seen that the frame rests upon a subbase which carries the main-bearing housings. Through bolts or tension rods connect the top of the frame with the subbase. The box frame itself is thus relieved of the working stresses. The working cylinders and the air compressor are held to the base by studs. Doors opposite each crank on both sides of the frame give access to the working parts. Small inspection ports allow the operator to observe the working condition of the bearings while the machine is running.

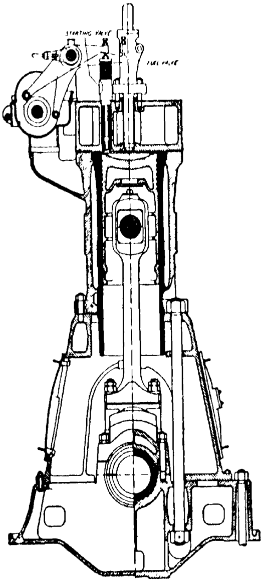


Fig. 69A.—Busch-Sulzer Type *B* frame.

Busch-Sulzer Type-C frame.—From 750 hp. upward the Busch-Sulzer Co. builds the engine on the two-stroke cycle principle, Fig. 70.

The frame, which in the four-cylinder unit consists of two boxes bolted together by means of a distance hood, Fig. 15, is bolted to the subbase. The subbase is in two sections to allow ease in handling, etc., and these sections are held together by finished bolts in reamed holes. The bearings rest in the cross girders of the subbase.

Since the crosshead type of piston is used, the frame is of some height. The crosshead guides are bolted to the cross webs of the frame. The large side doors make the frame probably the most accessible of any of the large engines.

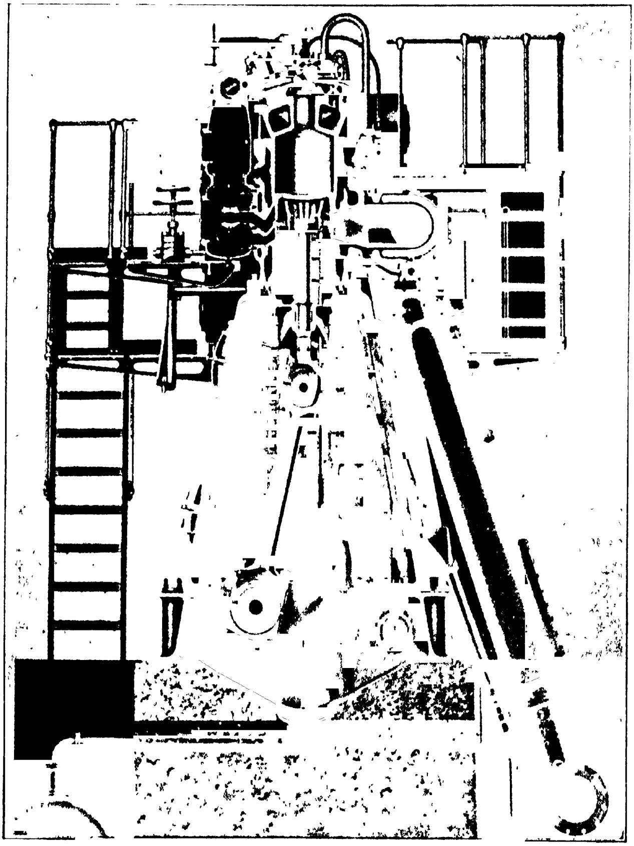


FIG. 70.—Cross-section of Busch-Sulzer Type C Diesel.

Standard Diesel Frame.—The Hadfield-Penfield Steel Co, uses a box frame for the Standard vertical engine. A cross-section of the base appears in Fig. 70 *A*, while Fig. 33 is a view of the engine assembled. The cylinders, two or more, are bolted to the box which in turn is bolted to the subbase. Tension rods carry the working load. The main bearing housings are in the cross girders of the subbase.

Winton Diesel Frame.—The Winton Diesel has a box frame which embodies no unusual feature. Fig. 71 is a cross-section, showing the method of bearing and cylinder support.

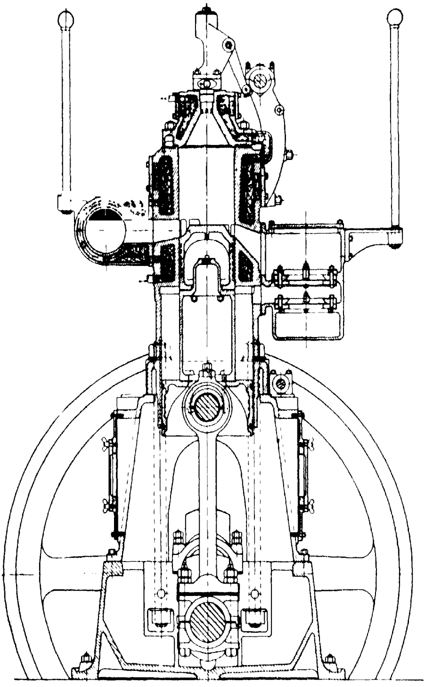


Fig. 70A.—Cross-section of Standard engine.

Lombard Diesel Frame.—The Lombard Diesel has a box frame upon which are mounted the cylinders, which are cast in one piece with the head and cooling jacket. The box frame rests upon a subbase carrying the shaft bearings. See Fig. 71*A*.

New London Ship & Engine Diesel Frame.—The New London Ship & Engine Diesel is primarily a marine engine, and in consequence the box frame is made as narrow as possible. An iron subbase carries the box frame. A view of the engine appears in Fig. 72.

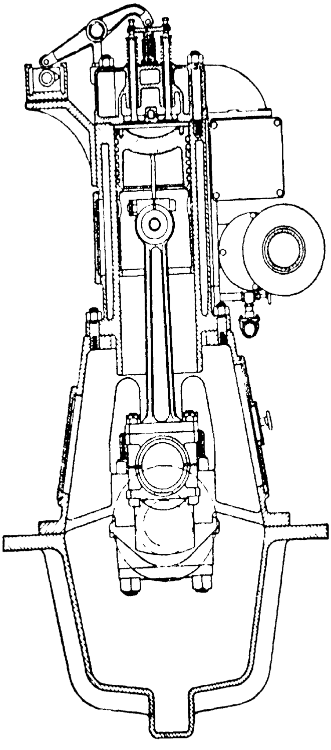


FIG. 71.—Winton frame

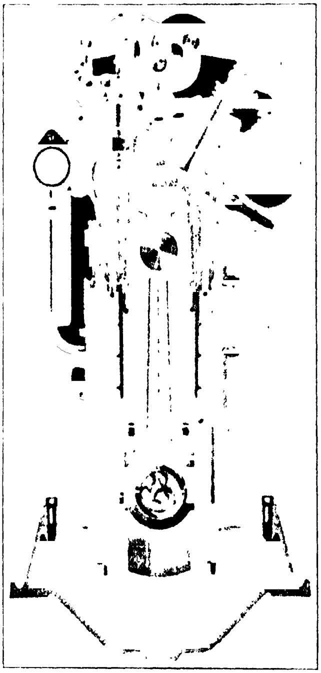


FIG. 71A.—Cross-section of Lombard Diesel.

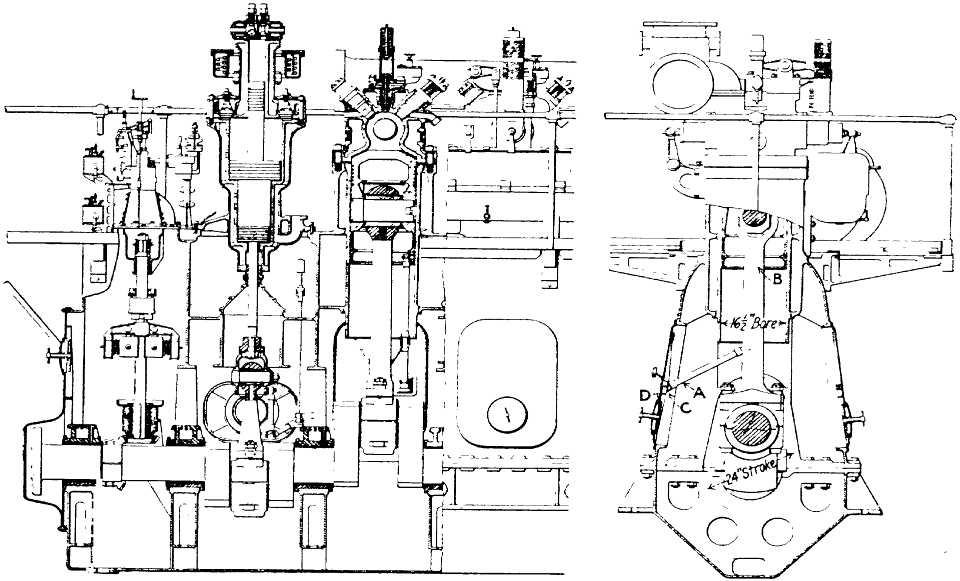


FIG. 72 —Cross-section of New London 600 hp engine

TRUSS FRAMES

Nordberg F. H. Diesel Frame. The larger Nordberg Diesels (550 hp. per cylinder) are provided with the girder-type frame, Fig. 72 *A*. In this design two heavy cast-iron vertical struts are bolted to the subbase and support a working cylinder. The crosshead guides are separate strips bolted to the girders. This

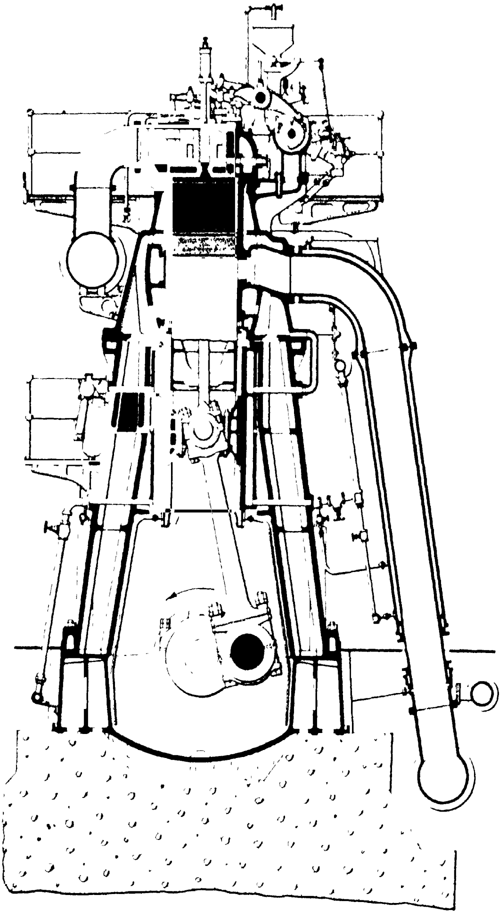


FIG. 72*A*.—Cylinder and frame of Nordberg *F D* Diesel.

engine of 550 hp. per cylinder has the largest output per cylinder of any Diesel of United States manufacture, and as a consequence the girders are exceedingly heavy.

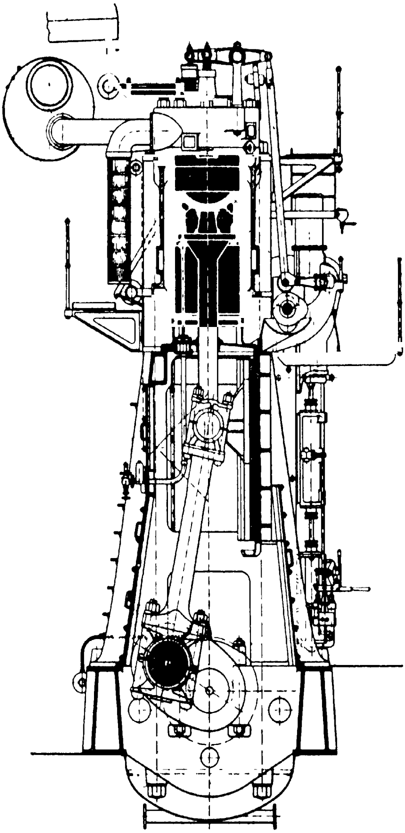


FIG. 73—Design of Cramp's B. & W. engine.

Wm. Cramp & Sons Diesel Frame.—The B. & W. Marine Diesel manufactured at the Cramp shops has the standard B. & W. girder frame, as shown in Fig. 73, which rests upon the cast-iron subbase. Each frame consists of a vertical column on each side of the shaft tied together by suitable cross-members. Upon the top of this frame are placed two distance pieces or lugs.

Each cylinder then rests upon four of these lugs; that is, two of the frames are required to carry a cylinder. The inside frames carry the lugs for two cylinders. In this way a six-cylinder engine needs eight frames. The cylinders rest upon the lugs, through or tension rods reaching from the lower part of the subbase to the top of the cylinder casting. These rods transmit the load from the cylinder to subbase.

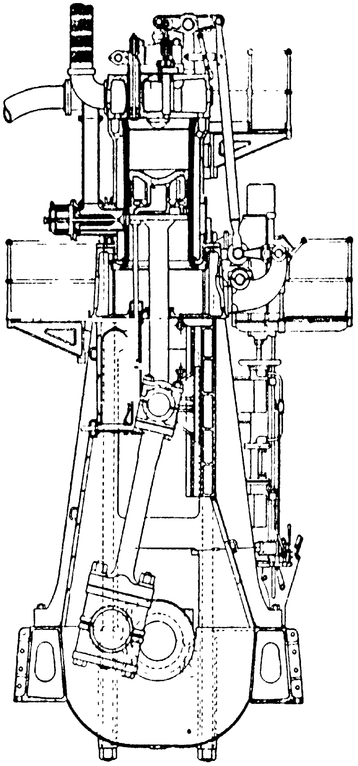


FIG. 74.—1,750 hp. Worthington frame.

Worthington Pump & Mach. Corp. Frame.—The 1,750 hp. marine Diesel built by this corporation uses the girder type frame, as shown in Fig. 74. Upon the top of this framing is set a heavy cast-iron box running lengthwise of the engine. The cylinders are seated upon the box and tension rods carry the load to the subbase.

This engine frame is peculiar in that the front frame legs are provided with gaps closed by a keyed wedge. This allows the shaft to be rolled out to one side for inspection without dismantling the entire unit.

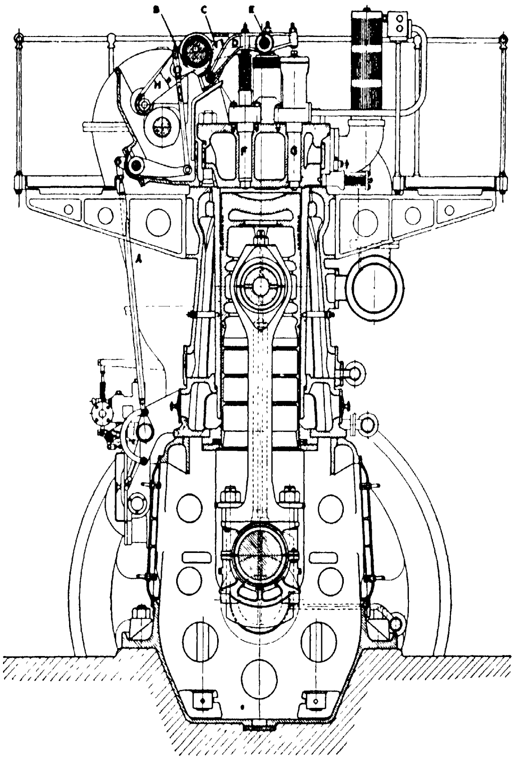


Fig. 75.—Snow 600 hp. vertical Diesel.

Snow Vertical Diesel.—Figure 75 shows a view of the cylinder and frame of the vertical four-stroke-cycle Diesel built at the Snow Works of the Worthington Pump & Machinery Corp. The frame is a one-piece box carrying cross girders into which the shaft bearings are set. The cylinders are bolted to the plane top of the frame. The bottom of the frame is made closed, and acts as a crankpit into which the lubricating oil drips after cooling the bearings and the pistons.

Werkspoor Frame.—Figure 36 illustrates the frame of the Werkspoor Diesel. It is quite different from the construction of any other engine. The cylinder head is cast in one piece with the liner. Three of these units are then set in the cast-iron box, the skirt of each liner being sealed at the lower end. This box serves as the cylinder jacket and in turn is bolted to the engine frame girders. This camshaft is driven by eccentric rods instead of through a vertical shaft and bevel gear. Crosshead pistons are used and being short are removable through the space between the lower end of the liner skirt and the top of the frame diaphragm.

CHAPTER VIII

CYLINDERS AND CYLINDER HEADS

Since the first Diesel engines followed gas engine practice as much as was possible, it is not surprising that the one-piece cylinder was incorporated in the first designs. At the present time no Diesel of any size is fitted with a cylinder having the liner cast integral with the jacket, though a few small units do use it on account of the smaller initial cost and the consequent lower replacement expense. For cylinders of 14 in. or less in bore the one-piece cylinder casting has no objectionable features. In fact, from the operator's viewpoint the one-piece cylinder is fully as serviceable as in the separate liner design. The cylinder walls are always amply thick for reboring.

American Diesel Engine.—This pioneer company employed a one-piece cylinder, as may be seen in Fig. 13. The bottom of the liner is not united with the jacket but is free to elongate without strain. This open end of the jacket cavity is closed with a cover ring which seats against a flat rubber gasket and is held by cap screws to the inner and outer wall ends. The top is not provided with cored water passages to the cylinder head, the water being passed into the head by outside gooseneck connections.

The lubrication of the cylinder is largely dependent on the splashing of oil from the enclosed crankcase. To make the oiling more certain, an oil line leads from a mechanical oil pump to the cylinder about midway down the cylinder.

American Cylinder Head.—This head, a cross-section of which appears in Figs. 13 and 76 is of irregular shape, one side carrying the cavity for the air admission valve. The exhaust-valve cage is some of the engines is bolted to the lower side of this projection while in others the cavity is cast with the cylinder body. The hot exhaust gases pass across the bridge or separator *A*, as does also the cold air charge. The alternate heating and chilling of this bridge produce shrinkage cracks that speedily extend entirely through the cast-iron wall. This allows the exhaust valve to leak. Many heads have been scrapped solely because of this bridge fracture. This is totally an uncalled-for extravagance since in all instances the fracture can be repaired by welding, or by the insertion of a wedge of steel.

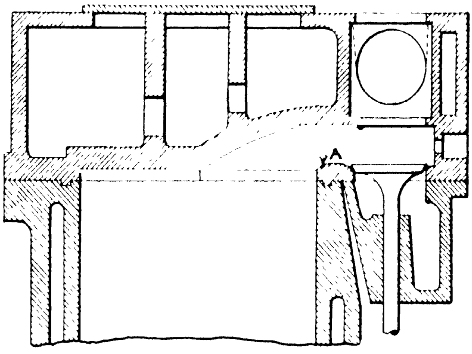


FIG. 76.—American Diesel cylinder head.

The head is fitted with a relief valve supposedly to blow at 800 lb. Unfortunately few operators test this valve; consequently it fails to function when an excessively high preignition pressure is experienced in the cylinder. The valve carbonizes and freezes to its seat unless relieved by being lifted at least once a week.

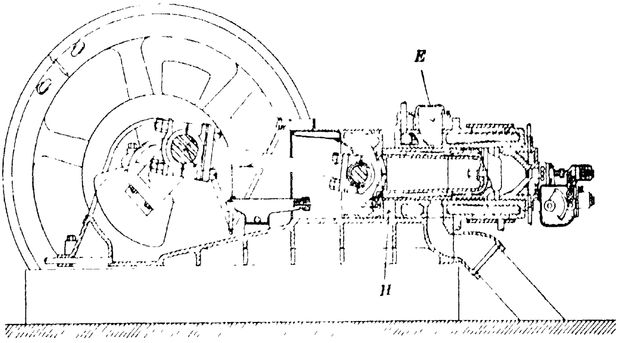


FIG. 77.—Cross-section of Standard horizontal Diesel.

Standard Fuel Oil Engine.—The cylinder of both the horizontal and vertical engines manufactured by the Standard Fuel Oil Engine Co. is of one-piece construction, when the power cylinder alone is considered. As outlined in Fig. 77, the liner and jacket are in one piece, the front end of which fits into the frame casting. The scavenging piston works in a bored cavity of the frame which is not provided with any means of cooling. The power cylinder, as mentioned above, is fitted into this cavity and is held by a flange. The cylinder casting is provided with ports, both for the exhaust gases and for the scavenging air. The air ports, at the top of the cylinder of the horizontal unit are arranged to give to the air charge a whirling motion which materially assists in the scavenging of the exhaust gases; see Fig. 78. The water spaces in the bridges are small and tend to scale if the water is bad since this is the place of greatest temperature. If the spaces once fill with scale, the bridges are subject to fracture. Periodical cleaning of the water jacket is imperative.

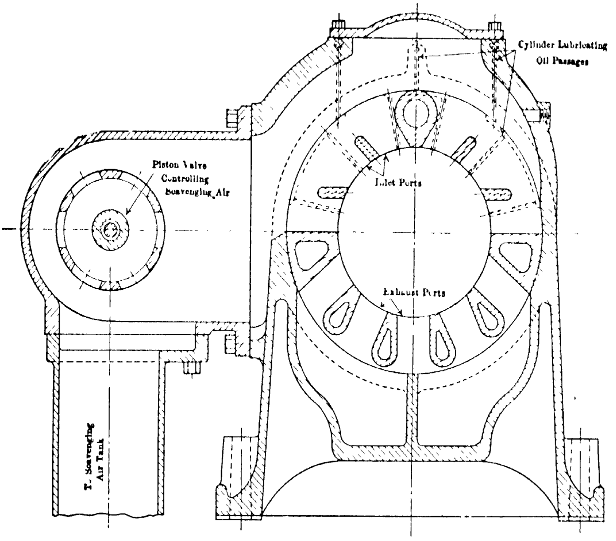

Cylinder Lubricating Oil Passages
Piston Valve Controlling Scavenging Air
Inlet Ports
Exhaust Ports
T. Scavenging Air Tank

FIG. 78.—Standard Diesel, section through air and exhaust ports.

Cylinder Head.—The cylinder head, Fig. 79, is water-cooled and contains but one opening—that for the fuel valve. Since the water line to the head is separate from the jacket cooling system, some engineers attempt to operate the engine on low loads with the head water line cut out. Since the head becomes hot, this does improve the combustion on low loads. However, there is danger in the liability of the head showing a fracture on cooling. The joint between the cylinder and the head should be metal to metal. In replacing the head, the surfaces require a thorough cleaning to avoid the risk of small particles injuring the gas-tight joint. In case of a damaged seat a thin copper gasket makes an excellent seat.

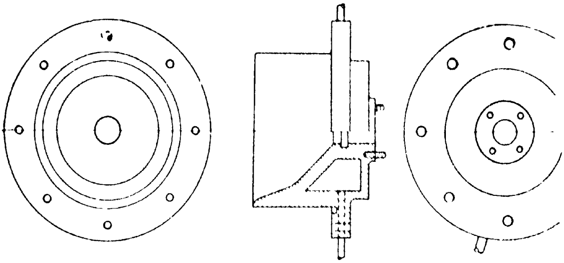


Fig. 79. Cylinder head of standard horizontal engine.

Busch-Sulzer Diesel Type B Engine.—This engine has the liner cast separately from the jacket, as appears in Fig. 80. The liner has a small flange at the top, which fits into a recess machined in the top jacket flange. The bottom is not anchored, being free to expand and contract. The head has four openings, for the exhaust, admission, fuel, and air valves. The Busch-Sulzer Co. has each cylinder head provided with a starting valve opening. This is used with two cylinders while on the remaining cylinders of the engine the opening is plugged. Figure 81 outlines the head of this engine, with the various openings as indicated. The same general lines are followed on practically all other vertical engines.

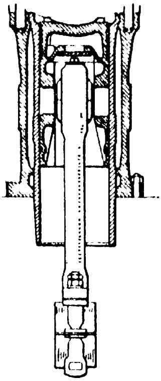


Fig. 80.—Cylinder of Busch-Sulzer Type B engine.

Busch-Sulzer Type C Diesel Cylinder.—The cylinder of the Type *C* two-cycle Diesel consists of a water-jacket casting and a liner. Since the engine is two-cycle the liner casting must carry a series of exhaust and scavenging ports, outlined in Figs. 70, and 82. In order to eliminate loss of air by blowing through into the exhaust, the scavenging ports are made in two series. The upper set of ports controlled by a rotary valve driven off the governor shaft. The exhaust port is longer than the lower air scavenging ports, with the result that these exhaust ports are uncovered, before the lower scavenging ports are uncovered. The burnt gases then rush out and the cylinder pressure drops to practically atmospheric before the lower scavenging ports are uncovered. The opening of these valves allow the scavenging air to blow into the cylinder, forcing the remainder of the gas out through the exhaust ports. Upon the reversal of the stroke the upward movement of the piston covers the lower set of air ports. The motion of the rotary air valve allows the air to continue to blow through the upper set of ports, scavenging the cylinder after the piston covers the exhaust port.

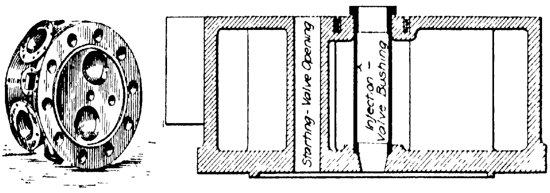


Fig. 81.—Cylinder head of Busch-Sulzer Type *B* Diesel.

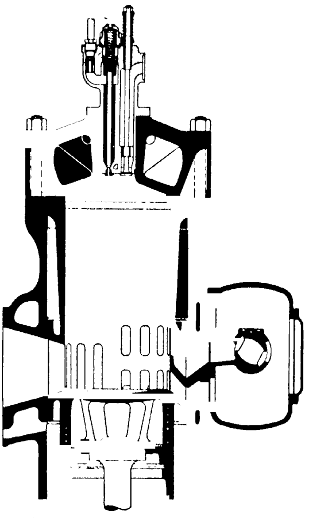


Fig. 82.—Cross-section Busch-Sulzer Type *C* cylinder.

This scavenging method enables the cylinder pressure to drop very low before the scavenging air blows in. The air pressure may then be very low —in some engines using port scavenging the air pressure must be high to prevent the gases in the cylinder from backing up into the air passages. The rotary valve not only allows thorough scavenging but likewise enables supercharging, or the filling of the cylinder with air above atmospheric, to be obtained.

In the Type *C* engine built since 1922 the rotary scavenging valve has been eliminated. As a substitute the upper scavenging ports are sealed by a set of shutter air valves of the same design as the valves used on the Type *C* scavenging pump. When the piston uncovers these ports the shutter valves are held closed by the cylinder gas pressure. As soon as the lower ports are exposed and the cylinder pressure has dropped to atmospheric, the shutters open and remain open until the upstroke of the piston raises the cylinder pressure to that of the scavenging air line whereupon the shutters close. This permits a slight surcharging and eliminates the rotary valve and its drive from the governor shaft.

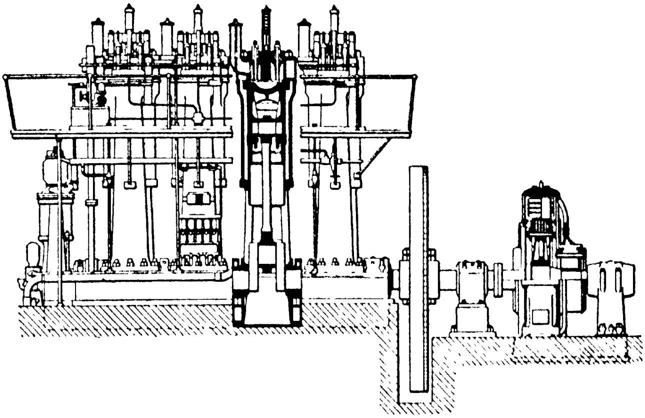


FIG. 83. McIntosh & Seymour A-frame Diesel.

The liner is pressed into the jacket, with a flange at the top and packing rings around the port belt and liner base.

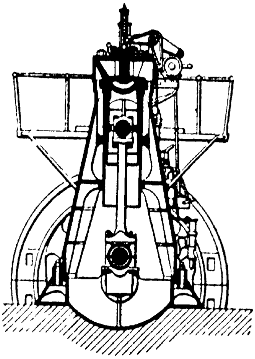


FIG. 84.—McIntosh & Seymour A-frame Diesel.

Type C Cylinder Head.—The cylinder head is a round symmetrical casting with one valve opening. This opening accommodates a casing which carries both the fuel valve and the air starting valve, Fig. 82.

McIntosh & Seymour Diesel Cylinder and Cylinder Head.—The McIntosh & Seymour Co. in its A-frame engines extends the frame to act as the cylinder jacket. The liner in many respects is the same as is found on a box-frame engine. This frame and cylinder construction is very popular with European manufacturers but is losing favor in the United States. It offers the serious objection of high replacement cost in event the cylinder-jacket wall fractures. This has happened on a few occasions in

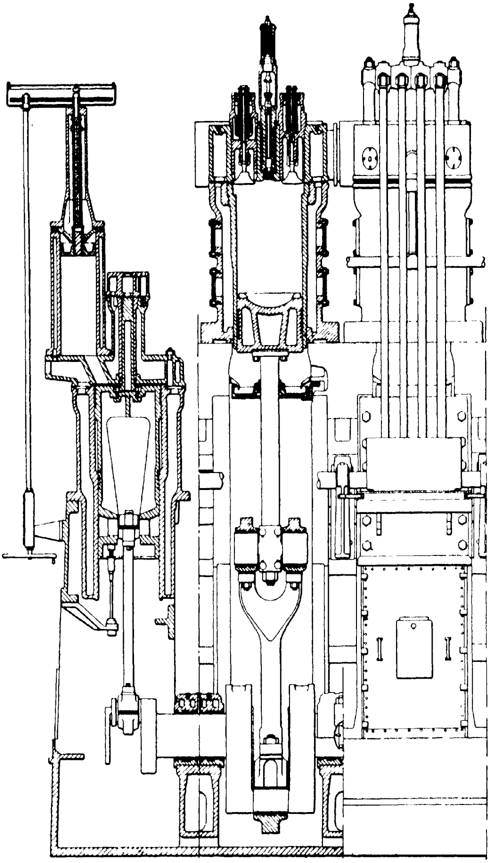


FIG. 85.—McIntosh & Seymour crosshead type engine.

this country and led to its abandonment by at least one manufacturer. Figures 83 and 84 illustrate the McIntosh & Seymour A-frame cylinder.

McIntosh & Seymour Box-frame Cylinder.—The Type B and trunk piston marine Diesels of the McIntosh & Seymour Co. have cylinders similar to that shown in Fig. 68. The crosshead type Diesel has a cylinder along the same lines as outlined in Fig. 85.

Cramp's B. & W. Cylinder.—The cylinder of the Burmeister & Wain Diesel as manufactured by the Cramp Co. follows the European B. & W. design. The cylinder is provided with a liner as shown in Fig. 73. The cylinder casting rests on distance pieces which in turn set on the frame standards. The cylinder heads are square rather than round and are bolted together in units of three. Long tension rods pass from the subbase to the top of the cylinder jacket, see Fig. 41.

One of the novelties of the engine is the removable partition placed on top of the frame standards below the cylinder. This partition is provided with a piston rod stuffing-box and prevents the leakage of any water into the crankcase.

Worthington Vertical Diesel Cylinder and Cylinder Head.—The vertical Worthington Diesel cylinder consists, as usual, of a liner pressed into a jacket. These cylinder castings carry lugs at the base, resting upon a box girder. The girder in turn is on top of the vertical frames. The use of the box girder affords a seal, separating the cylinder from the crankcase. In this manner no oil or water can enter the case from the cylinder. In addition the girder which carries three cylinders effectively ties the cylinders together, making a rigid connection. The cylinder is shown in Fig. 74.

The cylinder head is of the usual four-valve cavity casting.

Dow Diesel Cylinder and Cylinder Head.—The Dow Diesel uses the English Willan's Diesel design. The liner is pressed into the A-frame, being held by a flange at the top and rubber sealing rings at the lower end. The head follows modern practice, as revealed in Fig. 69. The casting has four valve cavities, the fuel valve being in the center, with the admission and exhaust valves at equal distances.

Winton Cylinder and Cylinder Head.—The cylinder of the Winton Diesel is somewhat unusual. The fact that the cylinder is small in size explains the design of a one-piece liner and jacket.

McIntosh & Seymour Diesel Cylinder and Cylinder Head.—The McIntosh & Seymour Co. in its A-frame engines extends the frame to act as the cylinder jacket. The liner in many respects is the same as is found on a box-frame engine. This frame and cylinder construction is very popular with European manufacturers but is losing favor in the United States. It offers the serious objection of high replacement cost in event the cylinder-jacket wall fractures. This has happened on a few occasions in

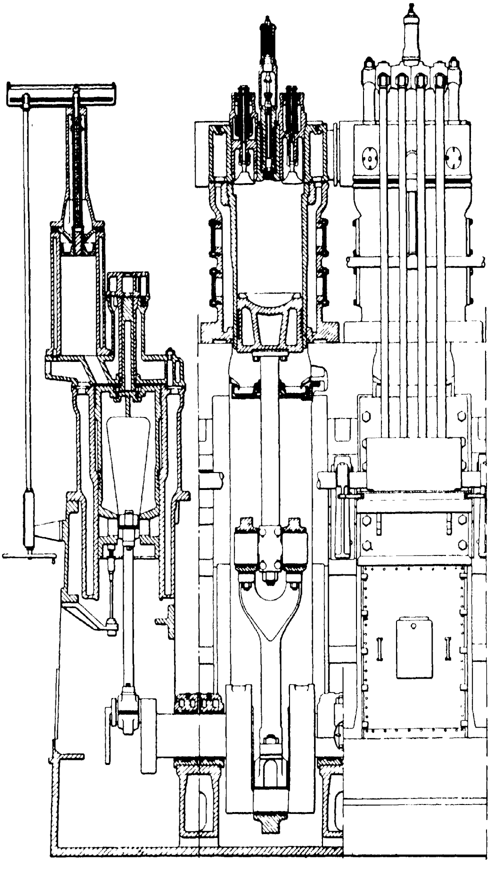


Fig. 85.—McIntosh & Seymour crosshead type engine.

The head is of standard construction. The valve cavities are separated from the water space of the cylinder head by thin thimbles instead of cast-iron walls. The thin-thimble walls transmit heat more readiy than the thicker iron walls, in this way reducing danger of fracture and overheating of the valve cages.

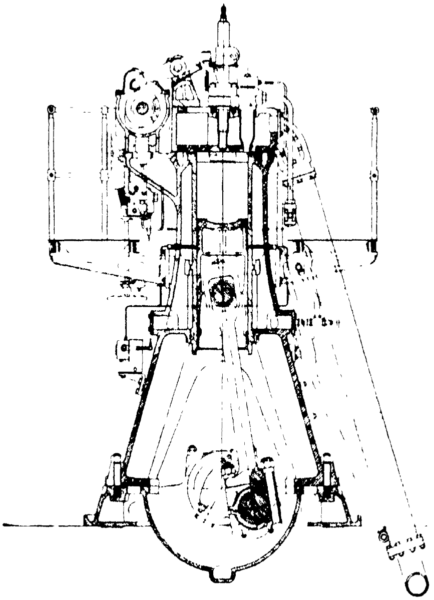


Fig. 86.—Cross-section of Fulton engine.

Lombard Diesel Cylinder and Cylinder Head—The Lombard Diesel has cylinders of a one-piece design, Fig. 71*A*. This casting also includes the head. Due to the arrangement of a separate skirt, the piston is easily removed from the bottom without disturbing the connecting rod.

It is the claim of this as of other manufacturers who use the integral head and cylinder that the danger of a fractured head is in direct ratio with the number of times the head is removed. The head bolts are seldom drawn up uniformily, and this, it is claimed, causes most fractures.

There are apparently certain advantages in this construction, but these are somewhat offset by the expense of replacement if the head does fracture.

Cramp's Diesel Cylinder and Cylinder Head.—The cylinder liner of the Cramp Marine Diesel is pressed into the jacket as shown in Fig. 73. The jacket is of a length such that an open space is provided between the frame top and the lower end of the liner. As a consequence any unburned fuel does not reach the crankcase. Long tension rods or bolts pass from the subbase up through the B-frames, ending at the upper part of the cylinder jacket. This is shown in Fig. 41.

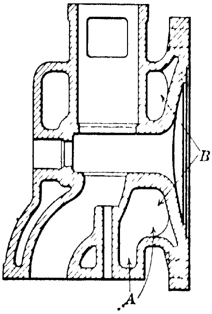


Fig. 87.—Cylinder head design with vertical valves.

Horizontal Diesel Cylinders.—Without exception all manufacturers of the horizontal four-stroke-cycle engine extend the frame casting to enclose the cylinder liner, thereby forming the water jacket without an additional casting.

Horizontal Diesel Cylinder Heads.—The manufacturers of the horizontal engine have two head designs from which to choose. If the head be of a symmetrical design, such as is found on the vertical Diesel engine, the valves must be placed horizontally. Figure 81 shows a very symmetrical casting that is closely followed on the Snow Oil Engine and on the De La Vergne F.D. Engine This design entails increased wear on the valve stems, and the seating of the valve is difficult. To obviate this condition many manufacturers have had recourse to a head with the valves placed vertically. This head, in order to keep down the compression volume, must be built somewhat along the lines of Fig. 87. It is apparent that this form of head will experience certain casting strains which will develop into fractures if they are not removed by the annealing of the entire head. If this procedure is followed, no great danger of fracture exists.

This design, however, does have one distinct advantage. In an engine having a flat cylinder head, such as, Fig. 81, and an almost flat piston head the clearance space consists of an extremely thin compartment of large diameter. The cooling effect is proportional to the exposed surface. On the other hand the design of the horizontal engine embodies a clearance space of some depth and fairly small diameter. The cooling loss is then less. In effect, the standard Diesel clearance requires a high compression ratio to secure the same compression temperature secured in the horizontal Diesel with a lower compression ratio.

In addition, when the fuel oil enters the usual combustion or clearance cavity it cannot contact with the air located along the outer portion of the narrow clearance space. Perfect combustion is somewhat difficult to attain. The oil spray also strikes the fairly cold iron walls, causing soot and carbon deposits. With the horizontal design the fuel blows into the mass of air which is in a deep cavity of small diameter. Combustion should then be easier to attain. The problem narrows down to a choice between better combustion and safer head construction.

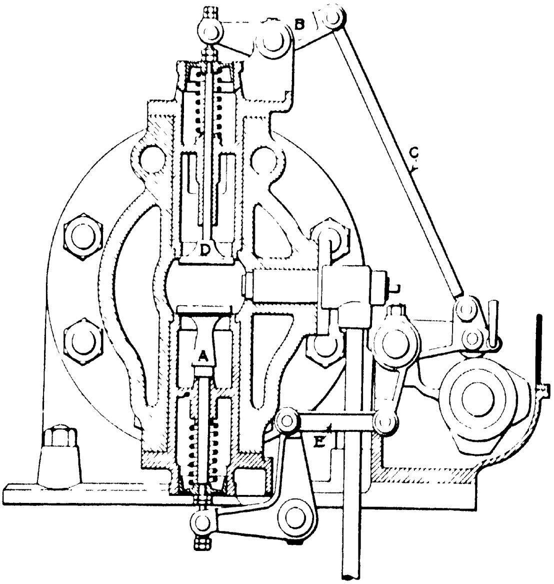


FIG. 88.—McEwen cylinder head

McEwen Diesel Cylinder Head.—Figure 88 is a view of the cylinder head of the McEwen Diesel.

Allis-Chalmers Diesel Cylinder.—Figure 89 gives a view of the cylinder of this engine.

Allis-Chalmers Cylinder Head.—The Allis-Chalmers Diesel employs the head shown in Fig. 90. The valves are set vertically while the fuel nozzle rests horizontally in the center of the head cover.

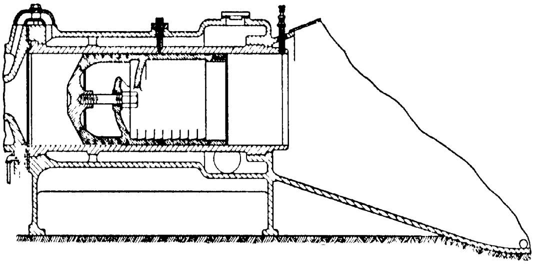


Fig. 89.—Allis-Chalmers cylinder.

National Transit Diesel Cylinder.—Figure 91 shows the cylinder and head of the first Diesels manufactured by this firm. Figure 92 is a view of the head adopted for the 1918 Diesels.

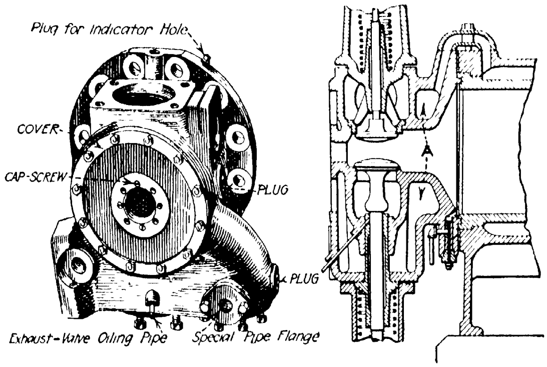


Fig. 90.—Allis-Chalmers cylinder head.

The valves are in a horizontal position while the head casting is simple, thereby removing practically all danger of fracture.

De La Vergne Diesel Cylinder and Head.—The Type F.D. Diesel has the cylinder liner pressed into the cylinder frame. The cylinder head is of symmetrical design carrying the valves in a horizontal position.

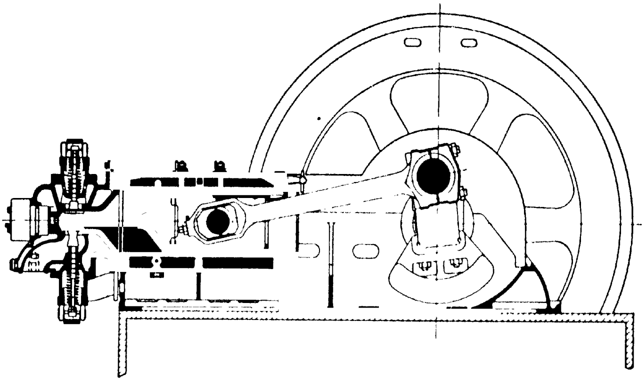


FIG. 91.—Cross-section of National Transit cylinder.

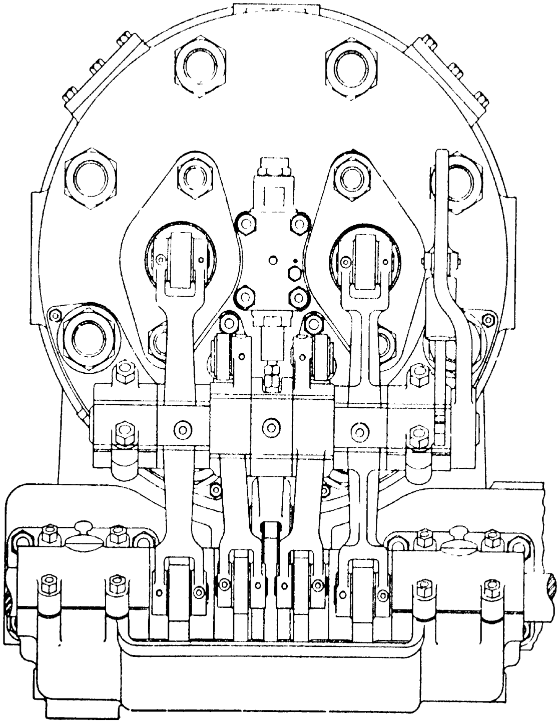


FIG. 92.—National Transit 1918 design Diesel valve rocker assembly.

Otto Diesel Cylinder and Head.—The few Otto Diesels built have had the liners pressed into the frame casting, while the head has the valves placed vertically.

Fractured Cylinders.—In cylinders having the liner cast in one-piece with the outer jacket there is a tendency for the liner to elongate more than the outer jacket wall since the temperature is much higher. If the cylinder is not designed with some means of allowing jacket elongation, a crack will eventually show up. In many cases the fracture will appear at the base of the jacket where the metal walls converge. Unless the cylinder is well designed there is danger.

This difficulty does not arise in cylinders with liners. Here, if fracture does occur due to other than cooling water troubles, it will be found that the damage may be traced to too thick a flange at the top. This will prevent the water from keeping the metal at a low temperature. In other cases, especially on large engines, fractures have been traced to excessively thick liner walls. This made the temperature gradient between the inner and outer walls too great with the result that the expansion of the liner created excessive tensile stresses in the outer skin, thus starting the break.

In high-speed, high-powered Diesels of the marine type the amount of heat generated per square foot of combustion space surface is large, as high as 600,000 B.t.u. per square foot per hour. A good deal of this is radiant heat which is absorbed by the iron walls. The rate of heat absorption is so high that enormous compressive stresses are set up in a thin skin of metal on the inner side of the liner walls. The skin tends to separate from the rest of the metal which is at a much lower temperature. Fractures thus created will soon go entirely through the liner.

These difficulties are present only in engines of large cylinder diameters or of high capacity, that is, with a high mean effective pressure. With the modern Diesel of medium speed and moderate rating these troubles seldom arise.

The difficulty of fractured cylinders has been largely eliminated as a better understanding of the necessity for proper cooling has come to the operating force. It can be safely stated that most cylinder fractures are traceable to improper cooling. Many plants follow a custom of cutting off the flow of cooling water as soon as the engine is shut down. Since there is about as much heat absorbed by the water as is given up in useful work, on shutting down a large quantity of heat remains in the iron parts—this must be taken up by the water contained in the cylinder jacket. This produces a rise in temperature sufficient to cause precipitation of the salts suspended in the water. These salts are deposited in the form of scale on the jacket walls. The action continues until the scale becomes so thick as to preclude the possibility of proper cooling. The cylinder walls attain a high temperature and develop fractures because of the inability of the red-hot walls to withstand the high cylinder pressure. Due attention to the cooling water will prevent any fracture in the cylinder liner.

Scored Cylinders.—In the chapter on pistons, several defects that would cause piston scoring are pointed out, and the discussion applies to cylinders. Where the scoring is purely local in character, the surface can be placed in working condition by rubbing with an emery stone, finishing up with a patient application of a scraper. Ordinarily, since the scoring is due to piston distortion, the defective surfaces are not in the plane of the crank and piston; consequently the reduction of the scored surface below the cylinder wall circle is not of any moment. Another type of scoring is at times encountered: this has the character of grooves and ridges due in the main to the side thrust of the piston under the high cylinder pressure. As long as the depth of the grooves is 0.005 in. or less no serious damage will occur. But when secondary ridges appear between the original ridges, the liner must be rebored.

Cylinder Wear.—It has been stated that the amount of liner wear in an accurately fitted oil engine operated under favorable conditions, should be at the rate of 0.001 in. per 1,000 hr. of service. As a basis for comparison, this figure is useless. It fails to specify the particular part of the liner subject to wear, and it is too low an estimate for the wear at the critical point, that is, just after firing takes place. Results on a large number of plants do not justify this generalization for any part of a liner surface. The wear varies enormously. A set of liners in the same engine, of the same age, exhibits different depths and rates of wear. For instance, the average rate of wear in thousandths of an inch in the case of four 12-in. liners of the same age in the same engine was found to be measured in thousandths of an inch as follows.

Parallel to the shaft near top	2 37	3 5	3 06	2 43
At right angles to shaft near top . .	2 25	3 43	2 93	3 06
Showing maximum variations up to	29 per cent and 51 per cent .			

And again:

Parallel to shaft at bottom of stroke	1 06	2 43	0 69	0 75
At right angles to shaft at bottom of stroke	1 25	1 43	0 88	0 75

showing even greater variations.

No definite inference as to rate of liner wear for the several parts of the wearing surface can be drawn. The wear should be gaged at intervals of say six months. When the compression becomes so low as to cause ignition failure or high fuel consumption, the liner should be rebored or renewed.

Reboring Cylinders.—The natural course of wear in the liner increases the clearances to such an extent as to require reboring. The liner, as an average, will need replacement or reboring every three to seven years, dependent on the hours of service and the attention it has received.

When the engineer is confronted with the problem of cylinder reboring, it is well for him to shift the work to the shoulders of some machine shop that makes a specialty of such work. The actual reboring is not hard, neither is the setting up of the boring machine; however, it requires a boring bar that will cost around $800, and few shops are willing to place their machine on a rental basis.

It is well, save in unusual cases, to secure a new liner from the builder as well as a new piston, if this is necessary. In many cases the piston may be turned down and a liner to fit purchased.

Liner Replacement.—All cylinders are of a thickness that will allow at least one reboring. If the liner becomes worn, after it has had one reboring, or if it is fractured, the withdrawal of the damaged liner is easily effected by the use of the draw-bolt, as outlined in Fig. 93. The spider may be made with either two or three fingers; the two fingers are as serviceable as the three fingers. The bolt is of 1⁹⁄₁₆-in. cold rolled shafting, the thread having a 1½-in. diameter. The spiders are placed over the cylinder flange and the front end of the liner as indicated. A part turn of the nut will bring the rod under tension; a few sharp blows on the inner surface of the liner at the crank end will, in most cases, loosen it so that the bolt can pull it out with ease. If the liner resists, additional bolt tension, followed with hammer blows along the liner supports, will expedite the removal.

In inserting a new liner the oil passages must check, as also must the dowels. After the casting has been driven into place by the use of a sledge and hardwood block, the bolt and spiders used in removing the discarded part can be reversed to press the liner into the recess.

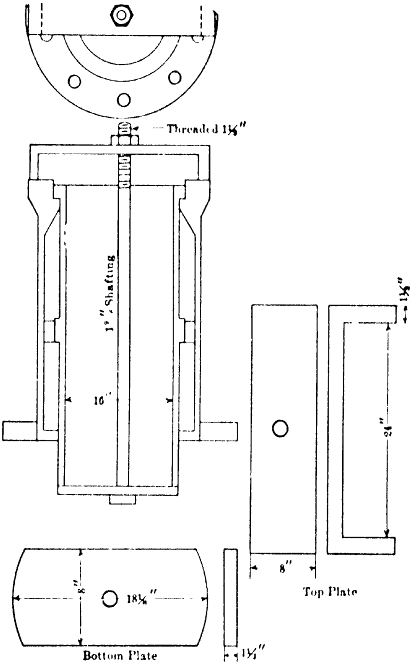


FIG. 93. Removing cylinder liner with screw jack.

Cylinder and Head Joints.—While a few engines depend on a metal-to-metal joint at the head to withstand the cylinder pressures, some form of gasketing is now well-nigh universal. The gasket may be either a flat copper ring, a copper wire, or a round rubber ring.

The flat copper ring is very successful as a gas check and is not difficult to make. Its objectionable feature is the large amount of sheet copper that is wasted in cutting the ring. Experience proves that the thinner copper sheet makes the best gaskets; $\frac{1}{32}$-in. thickness of metal is ample and enables the gasket to conform to the flange face. The gasket is best cut scant so that it fits easily into the gasket recess. If it is so wide as to require driving, the edges will bend and the gasket will not prove gas-tight. If a gasket cutter is not at hand, a pair of tinner's shears will be very satisfactory. A wooden mallet is handy to hammer the gasket to a flat surface.

In case sheet copper cannot be procured, an equally serviceable ring can be made of No. 10 gage soft copper wire; when the bare copper is not available, waterproofed electrical wire of No 10 gage may have its insulation burned off and the bare wire used. The wire is formed into a circle of the proper diameter and the ends soldered together. If the cylinder flange is not provided with a recess to receive the ring, the latter should be placed inside the bolt circle, touching each stud. This allows the leverage to be a minimum. The wire must be free from kinks or bends.

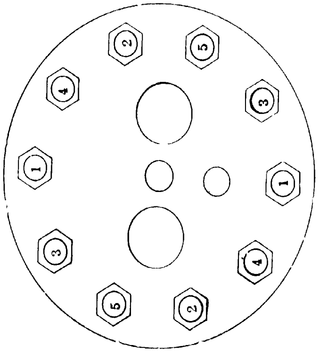


FIG. 94.—Method of drawing up head nuts.

Round rubber gaskets are often used on vertical engines, particularly on the McIntosh & Seymour engines. The rubber tubing is shaped into a ring of the proper diameter and the ends united by rubber glue. In engines where the cooling compartment of the head communicates with the cylinder jacket by cored openings at the flange the openings are surrounded with like tubes. These gaskets of rubber tubing can always be obtained from the engine builder, but any mill supply house will furnish the tubing in coils at a far less cost.

Drawing Up Cylinder Stud Nuts.—In tightening up the cylinder-head nuts, many engineers draw up one nut as snugly as possible before drawing up any of the other nuts. Such handiwork is evidence of a lack of mechanical knowledge and is to be shunned. If the top of the studs are numbered in pairs, similar to Fig. 94, and tightened in rotation, the head can be drawn down quite evenly. As example, all the nuts are run down against the head, then the No. 1 nuts are given an eighth turn, followed by a similar performance on No. 2 nuts, etc. Returning to the No. 1 nuts, they are given another eighth turn, etc. When giving the nuts the final movement, a workman can strike the wrench handle several sharp blows with a sledge.

Fractured Heads.—Fracture of a cylinder head is a malady which occasionally appears to afflict all makes of Diesels, no matter what the design may be. There is no doubt that the more complicated heads fracture often, but even the simplest of head castings do give way, usually on heavier loads. Bad water, without question, occasions the greater number of the fractured cylinder heads. This especially applies to the horizontal engine. Here, on shutting down, the water gives up its salts, which deposit on the iron surfaces. The greater part settles on the lower surfaces, marked *A* in Fig. 87. However, a part of the soft sludge adheres to the vertical surfaces *B*. This coating accumulates until it is as much as an inch thick. Since scale is an excellent non-conductor, no cooling effect is experienced on the hot cast-iron head wall which is in contact with the intense flame of the burning fuel. On cooling, the contraction of the iron gradually weakens the bond of the scale. This scale ultimately drops off while the engine is under load, exposing a red-hot iron surface to the cooling water. The sudden localized contraction of the iron on being chilled results in a fractured head. Evidently the horizontal head is more likely to shed the scale than is the vertical head. It becomes necessary for an engineer to inspect the cylinder head at stated intervals; if scale is present, it can be removed by scraping. If a solution of muriatic acid and water in the proportion of one to ten is allowed to remain in the jacket a few hours, all the scale can be washed out with a hose.

Heat stresses due to faulty design of the head also contribute to these fractures.

The combustion space in a Diesel engine is bounded by the cylinder cover, piston and the upper portion of the cylinder liner, or in the case of an opposed piston engine by two pistons and a portion of the cylinder liner. The temperature of combustion is is about 2,500 to 3,000 deg. F. and falls during the working stroke to about 1,200 deg. F., these figures varying, of course, according to the relative weight of the fuel and air charge. The heat flow through the material of the cylinder liners, piston and cylinder head, gives rise to stresses in the material which becomes more serious as the size of cylinder increases. The cylinder cover has always proved the greatest difficulty, as the casting is usually the most complicated and at the same time is subjected to the maximum temperature and pressure conditions. The cylinder cover and piston are together responsible for over 80 per cent of total heat flow to the cooling medium, the remainder of the heat being carried away through the cylinder liner. Although the water- or oil-cooled piston of a large engine carries away at least as much heat as the cover, the casting is much smaller than the cover casting, and freedom for expansion is more easily obtained. The liner of the normal type of engine is also a less difficult problem than the cover, as it is a perfectly symmetrical casting at the region of the combustion chamber, where the maximum temperatures are encountered.

Cylinder heads at times crack through the bridge between the exhaust and fuel valve cavities. These fractures are often due to improper cooling rather than to faulty design, although there are instances where casting fractures develop at these places as soon as the head is machined. To safeguard the engine head from such fractures that may develop in service, the flow of cooling water to the head must be positive and the temperature kept at a constant reading, the value of which can only be determined by experiment on the particular engine.

On marine Diesels the fracturing of cylinder heads is directly traceable to excessive overloads. In a heavy sea the propellor is at times exposed and immediately thereafter completely buried. This leads to engine hunting, and neither governor nor manual control can cope with the situation. The fuel pumps deliver excessive charges to the cylinder, creating pressures beyond the capacity of the heads to withstand.

Furthermore the marine engine, to reduce the weight per horsepower, is speeded far higher than the stationary Diesel. This applies especially to the submarine and light cruising engines. The heat absorbed per square inch of surface by the heads of a 500 hp. engine at 400 r.p.m. is double the amount absorbed by the heads of a 500 hp. at 200 r.p.m., since the cylinder bore area is practically one-half that of the latter engine and the total amount of heat absorbed is approximately the same. The cooling system, then, must be absolutely correct in design if fractures are to be avoided. The tendency of the salt water to scale is, of course, more pronounced where the temperatures are as high as exist in the marine Diesel heads.

Comparison of Two- and Four-stroke-cycle Cylinder Heads. It has been often claimed that since twice as many combustion events occur per minute in a two-stroke-cycle as in a four-stroke-cycle cylinder the heat stresses in the two-stroke-cycle head are much higher. This is partly true, but as an offset the two-stroke-cycle head is a more symmetrical casting and has a uniform cyclic temperature.

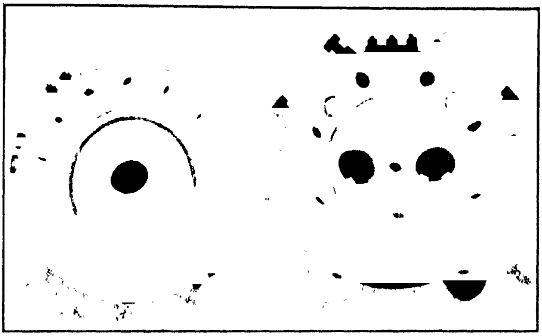


FIG. 95.—Two- and four-stroke cycle cylinder heads.

In the four-stroke-cycle engine the head has an exhaust valve cavity which absorbs much heat from the exhaust gases, and an admission valve cavity kept cool by the flow of air. This uneven temperature condition is the cause of head fracture. This fracture appears between the exhaust and fuel valve cavities, as in Fig. 95.

In the two-stroke-cycle engine with scavenging valves in the head there are no exhaust valves and the temperature is fairly uniform. Little danger of fracture from uneven temperatures exists. This is especially true in the case of the port scavenging engine where the head has only the one cavity for the fuel valve shown to the left in Fig. 95.

There are at least 1,500 two-stroke-cycle Diesel cylinders in service with the single fuel-valve opening in all parts of the world, and of these over 400 cylinders are above 20 in. in diameter. The mean indicated pressure is generally about 100 lb. per square inch and the power per cylinder in several cases is over 750 i.hp. If there was some defect inherent in the two-stroke-cycle design, as, for instance, if heat stresses had not been taken properly into account, there would surely be a considerable percentage of failures due to cracked heads. The fact that this has not been the case is the clearest proof that the design of the symmetrical head is sound both in theory and practice.

Another point of advantage associated with this design of two-stroke-cycle cover is that owing to the slightly concave shape of the under side, taken in conjunction with a similarly concave-shaped piston head, a well-shaped combustion space is obtained, contributing to efficient burning of the fuel.

Repair of Fractured Cylinder Heads.—Oxyacetylene welding has not been a marked success in the repairing of cracked cylinder heads. The cylinder head contains a great weight of iron, and, in welding, the flame is localized. The consequence is that the metal immediately about the fracture is highly heated and expands. After the molten metal is added, closing the fracture, the head is allowed to cool. The mass of the head has not been heated, and so shows no contraction. The obvious result is the shrinkage of the metal at the edges of the fracture, reestablishing the fault.

To weld a cylinder head successfully, a furnace can be constructed of fire brick. The floor should also be made of fire brick supported by old grate bars or iron rods. After a coke fire has been burning about the head for 24 hrs. the entire casting becomes red-hot. The oxy-acetylene flame is then applied; the fracture is enlarged to a trough shape, thus allowing the added metal to reach the bottom of the fracture. The new metal is deposited in small quantities and thoroughly welded to the cast iron before more is added. The head should then be left in the furnace to cool for 48 hrs. Since the entire furnace has been at a high temperature, the cooling will be very gradual, thereby avoiding all shrinkage strains. If the fracture has been across a valve seat, the part must be machined and the valve refitted. The process of welding here outlined has been followed with complete success, saving hundreds of dollars in a plant where five Diesels were installed.

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CHAPTER IX

PISTONS AND PISTON PINS

General.—The pistons of all engines properly fall into two general classes—Crosshead and Trunk Pistons. The crosshead piston is usually shorter than the trunk piston and is provided with a crosshead which receives the side thrust due to the angularity of the connecting rod. Figure 96 illustrates this type. With the trunk piston the upper end of the connecting rod is supported by the piston pin, which is fastened in the piston. Consequently the piston receives the side thrust which is taken up by the crosshead in the former type. The engine shown in Fig. 13 employs the trunk design of piston.

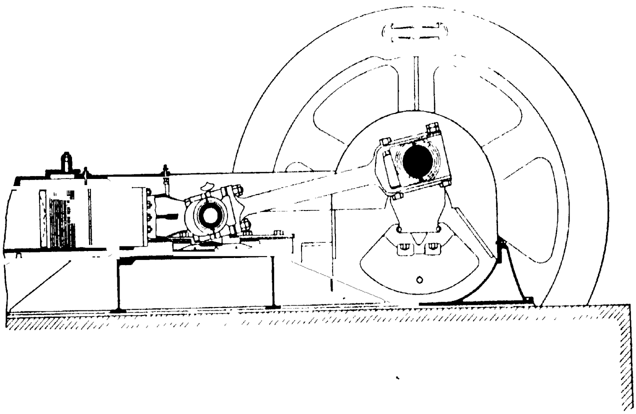


FIG. 96.—Cross-section of Snow Oil Engine employing a crosshead type piston.

The American manufacturers of small and medium-powered engines, up to 200 hp. per cylinder, have with few exceptions designed their engines with trunk pistons. Owing to the high cylinder pressures of the Diesel engine, the side thrust of the piston is of serious consequence, although in a 200-hp. cylinder the piston is usually constructed of a length sufficient to bring the side pressure within reasonable limits.

Trunk Pistons.—On units with a rating beyond 200 hp. per cylinder practically all builders use a crosshead design of piston; in fact, the first Diesel built, even though of 25 hp., employed a crosshead, and the present-day trunk piston is actually an adaptation of gas-engine practice. The trunk piston possesses certain features that make it attractive to the average operator. Since the side pressure is taken by the piston, there is no crosshead shoe to adjust. This adjustment, on a Diesel, must be made with a degree of knowledge possessed by none save experienced engineers. A guide clearance that would be quite satisfactory on a high-grade steam engine will prove entirely too liberal with the oil engine. This nicety of running fit necessitates careful adjusting of the crosshead shoes. The operator should understand that, beyond the pound that it occasions, a loose crosshead will allow the piston to bind in the cylinder, producing heavy scoring.

The trunk piston presents a problem in lubrication that does not exist with the crosshead type. The side thrust is borne by the trunk piston along its entire length, but this bearing surface extends over only a portion of the circumference. This rubbing area must be positively and copiously lubricated. The problem of lubricating a surface periodically exposed to hot cylinder gases is difficult. If the cylinder and piston are not oiled, either the piston or cylinder liner will cut.

Since the transverse pressure throws the piston against the bearing side of the cylinder, the clearance between the piston and cylinder must be less on the trunk than on the crosshead design. This is evident since, with the trunk piston, the entire clearance exists on the piston opposite to the wearing side. On the crosshead type the clearance is fairly well distributed around the piston; consequently a clearance between piston and cylinder of 0.007 in. is actually a clearance of 0.015 when the engine is firing. The crosshead piston can, then, be allowed a greater clearance than can the trunk type. This obviates danger of piston seizing when the engine is stopped after a run.

Crosshead Piston.—The crosshead design eliminates the heat difficulties of the piston-pin brass which are so often present with the trunk piston. Opportunity is also afforded for a heavier reinforced piston head. There is also less likelihood of the piston fracturing since it is not confined at the pin bosses. The crosshead and rod design admit of an oil guard at the front end of the cylinder thereby preventing the throwing of lubricating oil into the cylinder with the consequent carbonization. The dripping of dirty cylinder oil or unconsumed fuel oil into the crank case, where it renders unusable the bearing oil that is held there, is also eliminated by this design. To this can be ascribed the lower lubricating oil consumption of the crosshead design of engine. These manifest advantages are, it is the feeling of the majority of engine builders, offset by its greater complication of parts and the greater necessity for intelligent adjustments. As a consequence the small units are built with trunk pistons, going to the crosshead type when the cylinder sizes becomes so large that a trunk piston would require an exceedingly long cylinder. It is understood that the cylinder length depends upon the piston length which in turn depends upon the total cylinder pressure.

Some Engines had
only Oil Ring

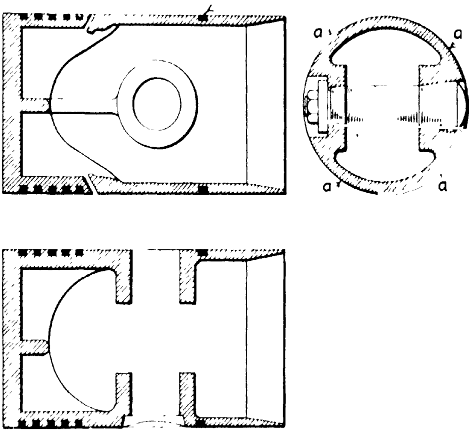


FIG. 97. American Diesel Engine Co piston.

American Diesel Piston and Pin.—This engine is, of course, no longer manufactured. However, many of these older engines are still in service, and the operator should be as interested in it. As in more modern designs the piston has a flat crown or top that is strengthened by ribs which extend down along the sides to the pin bosses, as shown in Fig. 97. The bosses are bored taper and offer support for both ends of the pin which are of different diameters. The pin is fastened into the piston by a washer and lock-nut; a dowel or short key at the large end prevents any turning of the pin. Since the latter is ground to a seat in the bosses, the contact is highly satisfactory.

The chief difficulty experienced with this design of piston is the distortion at the pin bosses, with a resultant oval shape. This distortion produces severe cutting on the piston sides along lines about 30 deg. from the pin axis. It is at once apparent that the pin holds the two bosses at a fixed distance; when the piston heats, the expansion will occur along the weakest section.

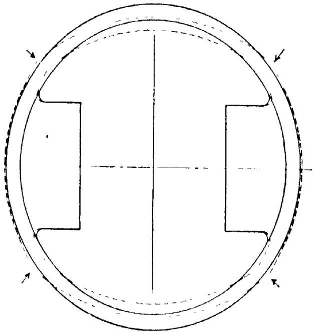


FIG. 98.—Piston distortion from elongation of the piston pin.

Figure 98 outlines the points *a,a*, of distortion. The pin bearing is oiled by the splash from the crankcase; consequently no oil passages are needed in the pin.

The number of rings used with this piston varies, ranging from four to seven. Frequently one of these pistons is found equipped with two rings per groove. In such event, the first step toward the elimination of ring trouble is the substitution of one broad ring in each groove.

Busch-Sulzer Type B Piston and Pin.—The Type B piston represents the modern development of the Diesel piston design. The head is thick to better resist the cylinder pressure and is concave. The clearance is small, and it has been necessary to cut away the edge of this concave to afford a clear passage for the gases as they enter or depart around the valves. To avoid distortion of the walls it is strongly reinforced by girth ribs. Seven rings are used about the upper part for sealing; a single ring is placed at the bottom as an oil wiper.

The chief variation from the usual American design is the water-cooling of the head. This feature is clearly shown in Fig. 99, although the water piping is not included in the drawing; they are the same type as used on the Busch-Sulzer Type C engine. The pipes are rigid and run parallel to the piston axis at each side of the connecting rod. The lower ends slide in the stuffing-boxes which are connected to the engine's water-piping system. This telescopic method of feeding the water is superior to the pantagraph or knuckle form. There is no inertia effect of swinging parts as with the latter type. The stuffing-box is much easier to maintain water-tight than is the knuckle. Even the telescopic arrangement will in time give trouble from weeping. In the Busch-Sulzer the crankcase receives all the return lubricating oil. For this reason leaks at the stuffing-boxes allow water to mix with the oil returns, practically destroying the lubricating qualities of the latter. On each occasion that the engine is stopped, the crank-case doors should be opened and the water connections examined.

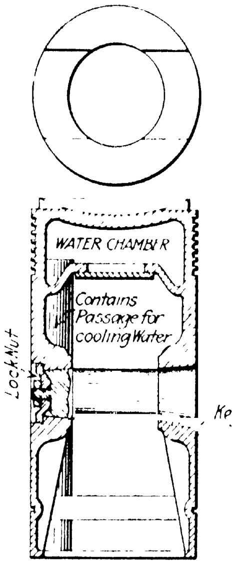


FIG. 99.—Busch-Sulzer Type B piston.

The piston pin is tapered at both ends and is held by a washer and nut as outlined. When the engine has been in careless hands, the pin may deform the bosses to such an extent that the pin has a poor bearing at the ends. It then becomes necessary to grind the pin to new seats. To accomplish this the dowel key at the big end must be removed to allow rotation of the pin.

Lubrication of the piston pin is effected through the hollow connecting rod from the crank bearing. Each time the hole in the rod registers with the oil hole in the crank pin oil flows up the rod.

McIntosh & Seymour Piston and Pin.—This company followed the designs of their Swedish associates in building the piston appearing in Fig. 100 for its trunk piston engine. The head is concave and is strongly reinforced on the lower side with a series

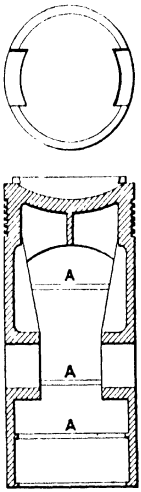


FIG. 100.—McIntosh & Seymour piston.

of ribs. These ribs do not extend down along the piston walls, which are supported by a set of girth ribs. The piston pin is straight and is held in bosses by set-screws.

The pin is lubricated by means of a slot cut in the piston surface. Oil from a timed lubricating pump enters this *V*-groove and flows down a drilled passage to a drilled hole in the pin and thence to the pin surface.

Fulton Iron Works Piston.—In Fig. 86, showing the engine cross-section, may be seen the type of piston used. The upper part of the piston cavity is closed off by a removable plate. This prevents oil striking the head and carbonizing. The top part of the piston of the larger engines is cast separate from the skirt as indicated in Fig. 101. Water cooling, by telescopic tubes, is found on engines with 20-in. or larger piston diameters. The one-piece piston of the small engine is shown in Fig. 102.

The piston pin has taper ends which are ground to fit in the piston bosses. The pin is driven into the bosses, and a nut and washer are placed on the small end as a safeguard against shifting of the pin.

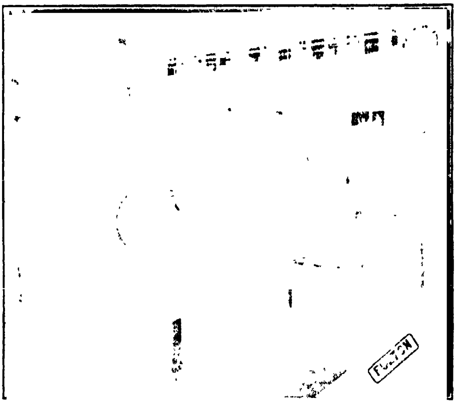


FIG. 101.—Fulton two-piece piston for large engines.

Lubrication of the piston pin is by means of a novel arrangement, Fig. 102. A tube *C* with a check valve *A* and spring *B* at its lower end extends downward from the piston-pin boss. A fixture *E* containing a hollow plunger with a check valve *D* is bolted to an angle iron on the engine frame. The lubricating oil line leads to this fixture. Upon each down stroke of the piston, *C*, the pipe to the pin, strikes the plunger *D*, and a stream of oil by reason of its inertia is forced into the pipe *C* and held by the check *A*. This device functions extremely well.

Dow Diesel Piston.—This Diesel piston is quite similar to other trunk pistons shown in this chapter.

Winton Diesel Piston.—The piston used in the Winton Diesel, appearing in Fig. 71, is unusual in one respect. The piston pin is clamped in the eye of the connecting rod and is supported at each end in a brass bearing fitted into the piston walls. There is, then, no adjustment for bearing wear. To remove the rod a wrench is passed through a slot in the piston, and the clamping bolt is unloosened.

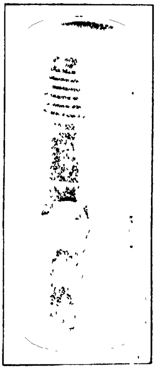


Fig. 102.—One-piece Fulton piston.

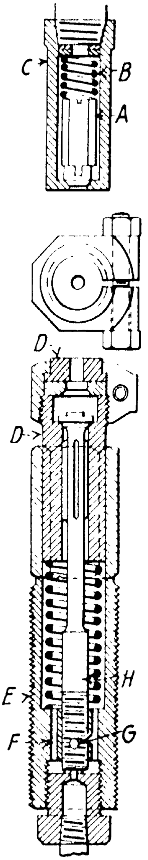


Fig. 103.—Fulton piston-pin oiling device.

Lombard Piston.—The piston of the Lombard Diesel is shown in Fig. 71*A*. It has no unusual feature other than the ribbing, which extends from crown to the bottom edge of the skirt. Oiling of the pin is by means of a tube from the crankpin bearing.

Allis-Chalmers Piston and Pin.—The first Diesel engines manufactured by the Allis-Chalmers Co. had pistons of the standard one-piece construction. The small clearance maintained between the piston and cylinder liner made this design impractical, and it was early replaced by the piston outlined in Fig. 104. The piston body is formed of high-grade cast iron and is provided with a false or removable head. This head is of nickel-steel, which develops fractures at a much slower rate than does cast iron, and is held in a machined recess by the stud shown. The piston head is conical in shape, and the impingement of the fuel charge is localized at the center of the nickel-steel head. This construction enables the builder to give a very small clearance between the piston and cylinder without danger of piston seizing due to head expansion. The compression is sealed by six rings while an oil-wiper ring is used at the base. To avoid the trouble of lubricating oil depositing on the inner side of the piston head and forming a hard scale, a baffle plate is incorporated in the casting. This plate is in two parts which allow a circulation of air. While this plate is fairly effectual in maintaining a clean head, oil will deposit on the inner walls of the piston, as in all horizontal engines.

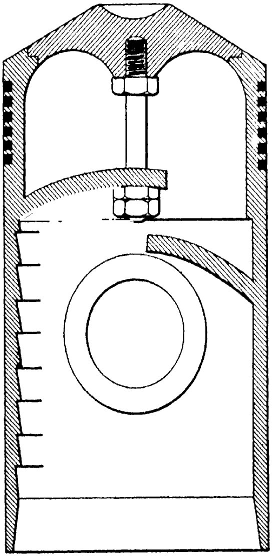


FIG. 104.—Section of Allis-Chalmers piston.

The piston or gudgeon pin is hardened and ground. The ends fit into straight bearings in the bosses, the piston being fastened by set-screws which are locked by smaller set-screws. The lubricating oil for the pin is deposited in a trough at the front edge of the piston, from whence it flows through a passage to the end of the pin. The pin is drilled on its upper surface to permit the oil to issue onto the bearing surface.

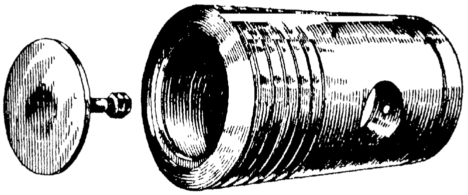


FIG. 105.—Allis-Chalmers piston.

McEwen Diesel Piston and Pin.—McEwen Bros. use a conical head piston of one-piece construction, Fig. 106. The head, after being cast, is annealed for the purpose of removing all casting strains. Six rings are employed to hold the compression while the wiper ring at the bottom has been dispensed with and a series of grooves made to replace this ring. The pin is held by a set-screw.

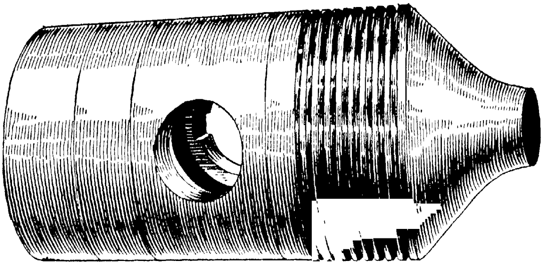


FIG 106.—McEwen piston.

Snow Diesel Piston.—The Snow horizontal engine has a piston that is a departure from the usual design with four-stroke-cycle engines. The piston, as seen in Fig. 107, is a single barrel casting with a separate steel head which is concave on its surface. The front end is bolted to the crosshead yoke. The crosshead is provided with a single shoe and has a wrist pin of extra large dimensions. There is a decided advantage in the crosshead design since the pin size is not restricted as it is in the trunk piston. In 1923 a one-piece piston was adopted.

In removing the piston from the cylinder, it is not necessary to dismantle the entire head and valve rigging. The connecting-rod can be unbolted from the crank pin, and the piston withdrawn through the frame. It is very essential with the crosshead piston that the shoe be properly adjusted. The engineer should measure the thickness of the shoe when first installed and endeavor to maintain this dimension by the insertion of shims.

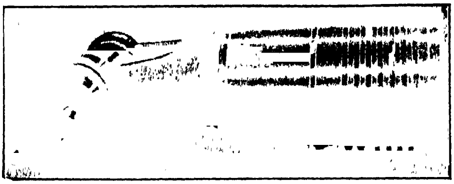


FIG. 107.—Snow cross-head type piston.

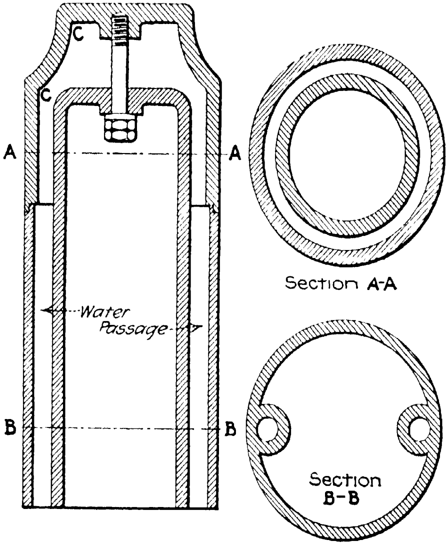


FIG. 108.—Standard power piston.

The Standard Fuel Oil Engine Piston.—The Standard horizontal Fuel Oil Engine is of the two-cycle design and has a stepped piston, outlined in Fig. 77. The main or power piston, of the horizontal engine Fig. 108, is a two-piece barrel casting.

The head is conical and extends down over the piston body forming a water-cooling compartment. The water lines are passages cored in the main piston casting and are connected to telescopic tubes at the front end. The power piston is bolted to the enlarged scavenging piston. The piston or gudgeon pin is bolted to this piston; consequently, the latter acts as the crosshead and receives the transverse thrust of the piston.

Since the crankcase does not act as the return lubricating oil receiver, leaks in the water tubes are not so serious as with some vertical engines. The water stuffing-box glands must be kept tight or the seeping water will destroy the lubrication on the walls of the scavenging piston. The cooling water discharge must be kept below 120 deg., and the line should be vented to avoid steam pocketing at the lower edge of the piston head, which receives a great amount of heat as the exhaust gases pass out through the ports.

The vertical Standard engine, as will be observed from Fig. 70 *A*, has a piston along similar lines, although water cooling of the piston is not used.

Worthington Marine Diesel Piston.—This engine, 1,750-b. hp. unit, uses a crosshead and fairly short piston, as shown in Fig. 74. Due to the size, piston cooling is necessary. Telescopic tubes provide the inlet and exit for the water.

Busch-Sulzer Type C Piston.—This two-stroke-cycle engine has a crosshead type piston, as indicated in Fig. 109. The piston proper is short, merely long enough to accommodate the piston rings, as all guiding is performed by the crossheads. It is provided with a water jacket immediately under its upper face. The piston rod is attached to the lower flange of this piston, by means of studs. The piston is of special, hard, close-grained cast iron to resist the heat and working conditions to which it is subjected, and is water-cooled by a patented arrangement. The water is injected into the cooling chamber in the piston head and conducted away by a system of telescopic tubes arranged on a principle which avoids all stuffing boxes or swing joints, while preventing oil and water leakage. These tubes are shown in Fig. 109 attached to the piston. Immediately below the piston proper is fitted a skirt, the sole function of which is to cover the scavenging and exhaust ports in the cylinder wall. The skirt is of hard, close-grained cast iron.

The piston rod is of forged open-hearth steel. The upper end is provided with an integral flange, for attachment to the piston, the lower end is forked, to connect to the crosshead pin.

The crosshead pin is a high-carbon open-hearth steel forging. The central part of the pin forms the bearing for the connecting rod. To each side of this bearing is bolted the forked end of the piston rod. The ends of the pin carry the crosshead shoes. Shims are provided between the crosshead pin and the end of the piston rod to permit the adjustment of the compression in the cylinder.

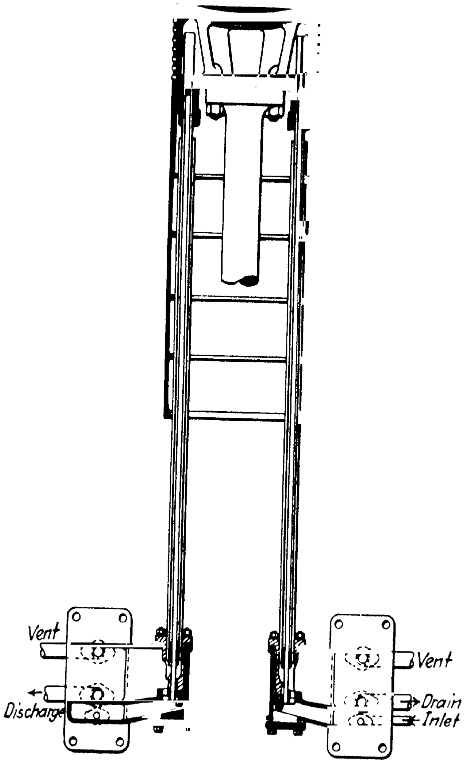


FIG. 109.—Busch-Sulzer Type *C* piston.

The crosshead shoes are of cast iron, with babbitted bearing faces and are of the double, central-guide type. No loose or adjustable pieces are attached to them; adjustment is provided by shims under the stationery guides which are bolted to the crankcase.

Nordberg Type V.E. Diesel Piston.—The piston of this two-stroke-cycle engine consists of a main portion or top which is water-cooled, and a lower barrel which acts as a seal to the scavenging and exhaust ports as shown in Fig. 68 *A*. The piston rod is bolted to the top piston and is likewise bolted to the cross-head casting. The water-cooling enters and leaves through the hollow piston rod. A hinged arrangement, Fig. 110, allows the water to reach the piston rod.

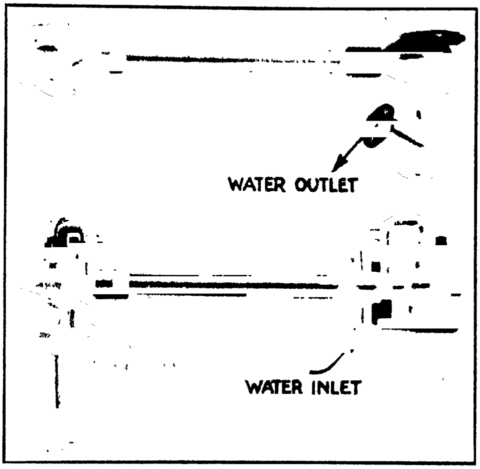


FIG. 110.—Piston-cooling lines, Nordberg engine.

The crosshead shoes are fitted with wiper blades which dip into oil troughs at bottom deadcenter, thus oiling the crosshead guides, which are of the bored type. Having port scavenging, the piston head carries a deflector lug.

The Type F. D. Nordberg Diesel has a piston built along similar lines, being illustrated in 72 *A*. The upper part of the piston is separate from the skirt and the piston rod is bolted to this upper section in a manner somewhat similar to that of the V.E engine. The piston rod is bolted to the crosshead pin which is fastened to the crosshead shoes as shown in Fig. 72 *A*. Adjustment of the shoes is obtained by the insertion of shims between the square side of the pin and the shoe. Piston cooling is through a trombone connection and the hollow piston rod, quite similar to that of the small Nordberg Diesels.

Worthington-Snow Diesel Piston.—The vertical Snow engine has a trunk piston as indicated in Fig. 111. It is unusual in that the 500- and 750-hp. units have the pistons cooled by oil. The lubricating oil is fed to the bearings by pressure feed. The oil passing through the drilled shaft and crank flows upward to the piston pin. It not only lubricates this pin but is forced through a passage into the piston crown. After picking up heat from the head the oil drops down into the crankcase. It is then pumped through a filter and cooler and is again forced through the bearings, etc.

One unusual feature of this engine is the stuffing-box placed at the top of the frame and immediately below the cylinder liner. By this arrangement any unconsumed fuel flowing down past the piston is caught and so prevented from entering the crankcase and mingling with the lubricating oil.

The piston pin has one end tapered and ground to fit the piston boss, being held into place by a washer and nut.

New London Piston.—The piston of the New London Diesel, which is shown in Fig. 72, is of the trunk type and has the upper end shut off by a removable plate. The crown is conical and since the clearance or combustion cavity is in the cylinder head the piston has but a small clearance at upper dead center.

The piston pin is lubricated through the hollow connecting rod.

Cramp's Diesel Piston.—The piston of the Cramp engine is very short being of the crosshead type, as may be seen from Fig. 73. The cooling of this piston is obtained by oil, which passes through a telescopic pipe to and from the piston. Unlike most engines, the piston may be removed by lowering it through the frame. The partition at the top of the frame is in two parts so that it is easily removed. In this way it is not necessary to lift off the cylinder head when the piston requires inspection or cleaning. The admission and exhaust valve cages are lifted out of the head and by inserting eye-bolts in the piston the latter may be lowered.

The crosshead is of the single-slipper type, the slipper being bolted to the crosshead pin. The pin has a square center portion through which the threaded lower end of the piston rod passes

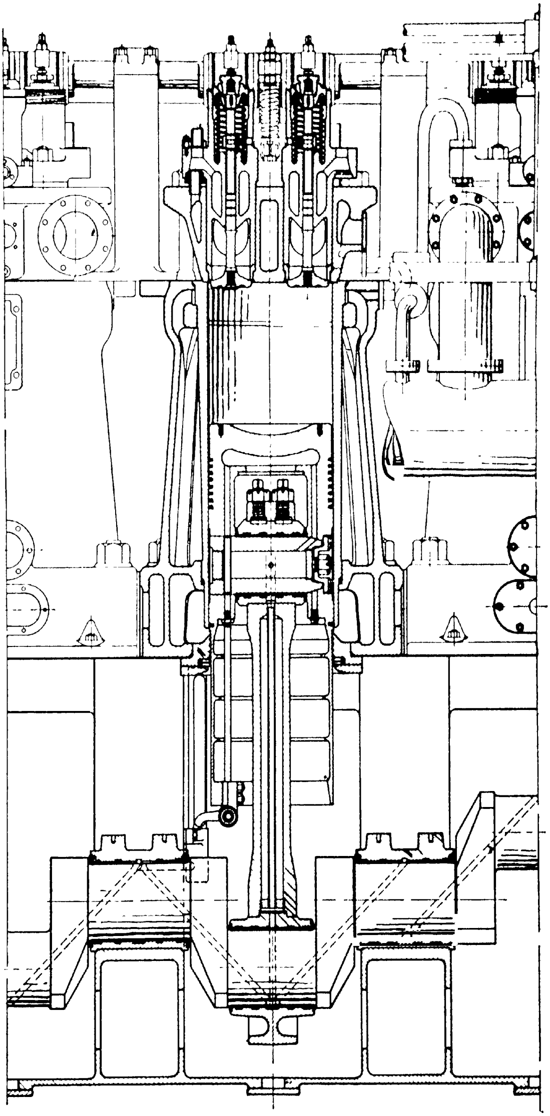


FIG. 111.—Snow vertical engine showing oil-cooled piston.

and is locked by a nut at the bottom. The connecting rod forked upper end engages the outer ends of the crosshead pin block.

Stuck Pistons.—It is by no means uncommon to find that on a Diesel engine one or more pistons stick or freeze to the cylinder walls. This is traceable to three or four causes.

In the first place some cast irons have a marked tendency to "grow" or increase in diameter. As a result, the piston in time increases in diameter until close metallic contact is made with the cylinder liner. The engine loses power and finally stops. The freezing becomes so intense that the piston may be removed with only the greatest effort.

This growth apparently is due to the iron having a low elastic limit under the high cylinder temperatures. The pressure then distorts the piston which takes a permanent set since it has been stressed beyond its elastic limit.

To correct this the piston should be tapered from the top extending down to the first ring. The clearance at the edge of the top may well be as much as $\frac{1}{16}$ in. Some builders always taper the piston and have the straight portion of the piston extend only over the middle third of the piston length.

The second and more common cause of seizing or freezing of the piston is lack of cooling water. This may occur both while the engine is in operation and after it is shut down. The former is evidenced by the loss of power and decrease in the engine speed and usually happens on a change from light to heavy load. The average operator understands that on full load the quantity of cooling water required is greater than on the lower loads. As the load comes on, the usual practice is to increase the flow of the cooling water. This chills the cylinder liner and causes it to contract before the hot piston experiences any effect from this additional cooling medium. This contraction lessens the working clearance; the piston becomes hotter and ultimately grips the liner walls. The remedy is to decrease the amount of cooling water as the heavy load comes on; then, after the cylinder liner warms up, the flow of water can be gradually increased while the discharge temperature is kept fairly constant.

There are, also, occasions when the seizing can be attributed to a hot piston pin, and ordinarily this is noticeable on starting the engine. After an engine has been in operation for some minutes, the heat absorbed by the piston is equaled by the heat given off by the piston to the cylinder walls, etc. The heat contained in the piston when the engine is shut down is large. If the flow of cooling water is discontinued at once, this heat is slowly radiated and a great part is absorbed by the piston pin. The pin elongates as a result of this heat absorption. The end thrust of the pin produces a change in the shape of the piston walls; especially is this true when the walls are not strongly ribbed. The deformation of the piston occurs along the thinnest sections, which commonly are at the junction of the pin bosses with the piston walls. These ridges or deformations produce severe cutting of the cylinder walls. Many liners are ruined by this action. When this misfortune is experienced, the sole relief other than piston replacement is the filing of the high spots until the piston is again cylindrical. The surface can then be dressed by emery cloth. Filing will also smoothen the scored cylinder walls. A quick and efficacious repair can be accomplished by first dressing down the rough places with a fine emery wheel, say 80J grade, held in the hand, finishing with a file and emery cloth. To some this probably seems a radical treatment, but extensive experience on many scored pistons and cylinders tends to prove that this is an effective way to rectify the damage. If the engineer is careful in using the emery cloth, the cylinder and piston can be made as smooth as when new.

Piston seizing is at times in evidence when turning the engine over at the beginning of a run. If, at the close of the last run, the cooling water was shut off too early, the heat in the piston head may evaporate all the lubricant on the piston pin. As the dry bronze bearing absorbs the heat, it cannot expand outward because of the greater mass of the connecting-rod end. The bearing closes in on the pin, which is also expanding; this action results in a gripping of the pin that is not completely loosened even after the parts have cooled. On starting, this wedged bearing restrains the motion of the connecting rod, and the engineer calls it a seized piston, though it actually is a case of a seized piston pin.

Still another cause of freezing is the carrying of too low a compression pressure. This in turn lowers the temperature at the end of the compression stroke. This temperature is not high enough to ignite the heavier part of the fuel. The heavy portion settles on the piston and cylinder walls. The rings gum, and eventually the piston sticks. The remedy is to raise the compression pressure.

Cooling of Pistons.—One of the main problems of Diesel design is that of getting rid of the heat that is absorbed by those parts about the combustion chamber. The jacket-cooling water as well as the head-cooling water removes the heat absorbed by these parts. On the other hand, the fuel is injected downward toward the piston head and the temperature at the head is much higher than at the cylinder wall or cylinder head. The removal of this heat is necessary if piston fracture is to be avoided.

On small engines, with pistons of 18 in. and less, the natural air circulation in and out of the piston usually is all that is needed. On large cylinders some form of forced cooling is required. It was formerly the practice to use pure water only, even in the case of marine engines. This has been changed and now sea water is used without damage.

In the case of engines in which sea water is used for cooling the pistons, a small header branches off from the main supply pipe and it in turn has branches leading into each piston. The return water from each piston flows into a header, and this header discharges into the engine-room bilge. The reason this system discharges into the bilge is that the discharge from each piston is arranged to flow into a funnel on the discharge header so that the operator may observe the flow from each piston and test the temperature by feeling, consequently, only atmospheric pressure can be carried on this line.

On Marine engines whose pistons are cooled with fresh water or oil, the system is a closed one and positive circulation is secured by means of a separate pump. A cooler is placed in the circuit so that the water or oil that is heated inside the pistons may be cooled before it is again pumped through the system. A variation in the case of oil-cooled pistons is to discharge the hot oil from the pistons into the crankpits, where it mingles with the lubricating oil and passes into the sump tank.

Lubricating Oil for Piston Cooling.—Formerly, oil was the only cooling agent used in pistons, the reason for this being the difficulties experienced in preventing leakage of the cooling fluid into the crankpins and into the lubricating oil. Practically all lubricating oils available were compounded oils, and water leaking into the lubricating system formed an emulsion from which it was very difficult to separate the water. This emulsion quickly clogged the lubricating system. This led to the simple expedient of using a portion of the lubricating oil to cool the pistons. This method is in use on some trunk-piston engines using forced lubrication, a portion of the oil forced to the piston pin being led through a passage into the piston and after passing through the piston, being discharged into the crankpit. The objections to this system are that the heat conductivity of the oil is rather poor, the heat of the piston sometimes causes the oil to carbonize and the carbon impairs the lubricating qualities of the oil and clogs the passages in the system, and the supply of oil to the pistons may become insufficient, owing to loss of oil through the ends of bearings that have too much clearance.

The operator of engines using this method of cooling should be constantly on guard against trouble from these last two conditions. The author has experienced persistent bearing trouble with a certain make of engine and this trouble was finally traced to the clogging of oil passages by flakes of carbon formed on the inner side of the piston top and flushed into the lubricating system. In the case of a group of engines of another make constant trouble was experienced from cracked piston heads. The pistons were dished on top and the cracks invariably occurred in the center of this dished top, the metal cracking in much the same manner that muddy ground cracks when dried by the sun. Examination of the pistons revealed that the outlet pipes extended only a short distance into the piston so that the oil flowed out without filling the piston, allowing the portion of the piston head directly under the spray valve to become overheated. The outlet pipes in all the pistons were changed so that they extended into the pistons to a point close to the under side of the piston top. No oil could flow out until the piston was full and contact between the oil and piston top was assured. After this modification no more pistons cracked. In passing it may be remarked that these cracked pistons were made serviceable by boring out the cracked center of each of the piston tops and closing the hole by screwing in a plug and welding it in place.

Water for Piston Cooling.—Water is used almost universally for the cooling of land engine pistons. Since the specific heat of water is twice that of oil the amount to be pumped is one-half the equivalent amount of oil. If the operator keeps the water lines in shape there should be no leaks into the crankcase.

While water is used on marine Diesel pistons, even with a "straight" mineral-oil lubrication, difficulties are experienced if salt water is allowed to mix with the oil. The oil will form an emulsion with fresh water, but water can be easily settled out. For this reason some marine engines are equipped with fresh-water piston-cooling systems. Usually, one of the double-bottom tanks is used as a fresh-water reservoir, a special pump forces the water through the pistons, and after leaving the pistons it is passed through a cooler before being used again. This cooler is usually constructed on the surface-condenser principle, and salt sea water is circulated through it to cool the fresh water. For this reason the water in the piston-cooling reservoir should be tested periodically for chlorine. The presence of salt water, as indicated by an abnormal chlorine content, calls for an immediate examination and test of the piston-water cooler for leaks.

Piston Rings.—Piston rings are almost without exception simple cast-iron snap rings. Special forms of eccentric-turned, spring-controlled, and double rings have been tried but usually have been abandoned in favor of the ordinary rings.

In figuring the proportions of rings Reinhardt's formula of

$$\frac{r}{s} = \sqrt{\frac{k}{12p}} + 0.05$$

is often employed. Here

r = radius of cylinder.
s = ring thickness.
k = allowable ring stress, say 16,000 lb. per square inch.
p = ring pressures against the walls.

The width of the ring may be from 1 to 2 times the thickness.

The diameter of the ring before cutting should be about $\frac{3}{16}$ in. larger than the cylinder diameter. After cutting, the $\frac{3}{16}$-in. section should be removed, the ring ends pinned, and the ring turned to the cylinder diameter.

The customary designs of rings have lapped ends and are constrained from shifting in the groove by dowels. Some builders fasten the ring ends together with a pin. It is doubtful whether this serves any useful purpose. In turning up new rings, the casting should be made from gray cast iron free from scrap. The outside of the ring is machined to size. Then the casting is checked $\frac{1}{32}$ in. out of center and the inside turned, the ring next being cut off. This produces a ring of a varying thickness and give an almost uniform pressure entirely around its circumference. The outside diameter of the ring should be $\frac{3}{16}$ in. greater than the cylinder bore of the engine. In cutting the lap, the clearance is best made about $\frac{3}{16}$ in., while the lap may be as much as 1 in. The lap can be drilled and then cut out with a hack-saw, as shown in Fig. 112. The edges of the ring are left square and should never be rounded as is practiced by many engineers beyond removing the burr.

Fractured Piston Heads.—The cracks that develop in the piston head can be placed in two classes; namely, those which take a circular form and appear around the base of the concave portion of the head, and those fractures which are radial in direction. The former are caused by heat strains with a consequent breathing action at the base of the cone. These fractures seldom prove serious, and it is possible to continue to operate the engine without piston replacement. At times these fractures appear in conjunction with cracks on the interior side of the piston head, across the reinforcing ribs. In such cases the entire head may give way.

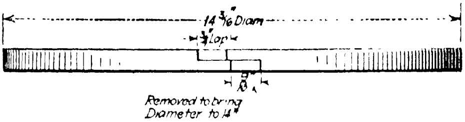


FIG. 112.—Piston ring lap.

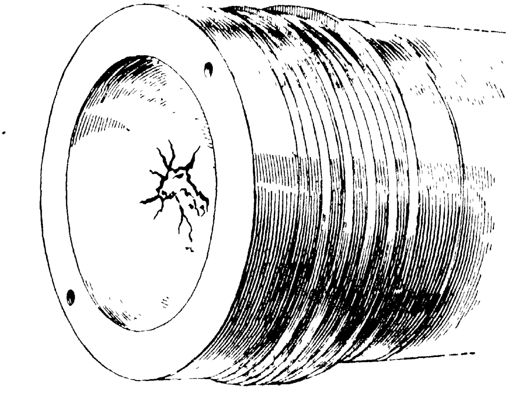


FIG. 113.—Fractured piston head.

The really serious fractures are those that develop radially in the head. Often these cracks extend across the head some 6

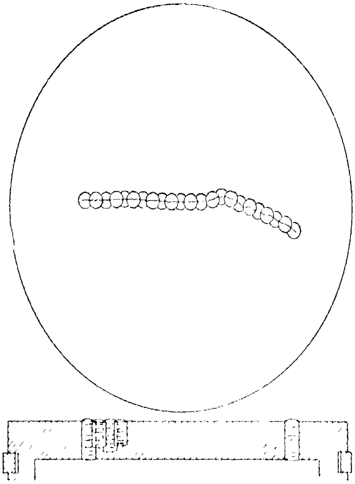


FIG. 114.—Sewing fractured piston head.

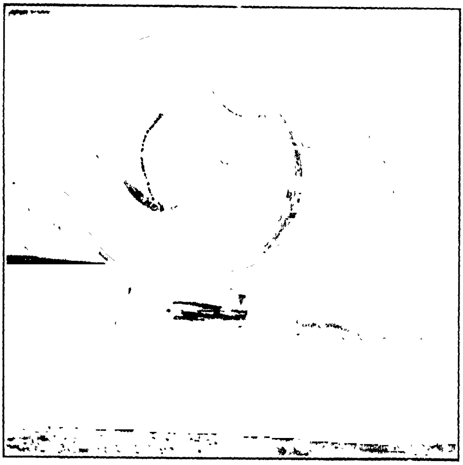


FIG. 115.—Sewed piston after 6 month's use.

to 10 in., Fig 113. The danger lies in the now non-rigid head allowing the piston to distort, scoring the cylinder walls. When the fracture is only a few inches in length it can be repaired by "sewing." A hole should be drilled at each end of the crack to prevent any further development. A series of $\frac{3}{8}$-in. holes are drilled and tapped along the line of the fracture. Into these holes threaded brass plugs should be inserted and cut smooth with the surface. Between these plugs a second row is inserted, lapping over the first plugs. This entire line of plugs is then hammered smooth; see Fig. 114. This "sewing" has been practiced with success on pistons as large as 18 in. in diameter. Figure 115 shows a piston six months after sewing. Welding if done correctly is a success.

Piston Clearance. No hard and fast rule can be given as to the allowable clearance between the piston and liner. It may be stated that generally this should be from 0.006 to 0.0012 in. on pistons of from 10 to 20 in. Much depends upon the design of piston and upon the cast iron of which it is made. Many show a tendency to grow at the crown and for this reason the upper end above the top ring should be tapered, giving a side clearance at the top of about 0.02 in.

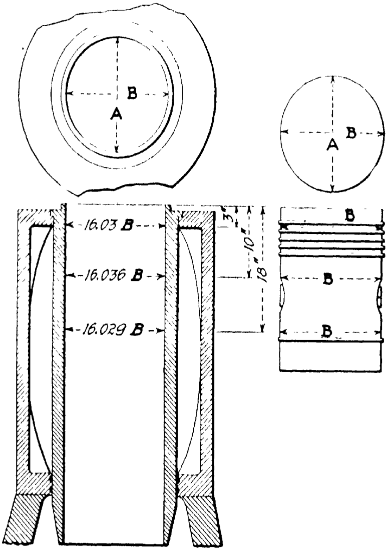
While the clearance on a new piston should be as stated, a piston can wear considerable before renewal and reboring or replacement of the liner becomes necessary. Engines have been run with clearances as great as $\frac{1}{8}$ in. Under such conditions there is a good deal of piston slap and the engine runs noisily. It may be taken as a general rule that for pistons up to 18 in. in diameter a clearance of 0.10 in. and for larger pistons a clearance of 0.08 in. is about as much as should exist before renewal. Greater wear will increase the fuel consumption and lower the capacity of the engine.

The rate of piston and liner wear varies, although the figures below give the wear on one engine.

LENGTH OF SERVICE, HOURS	RATE OF WEAR PER 1,000 HR., THOUSANDTHS OF AN INCH
4,000	2.8
16,000	2.3
21,000	2.8
25,000	3.5

The operator should check the wear at least every six months. If a drawing similar to Fig. 116 is kept on hand the wear may be recorded at each inspection.

To maintain the proper compression pressure as the wear increases the clearance between cylinder head and piston may be decreased by shims under the rod big end. It is likewise possible to secure correct combustion at low pressure by making the fuel admission earlier. There is danger in this, as excessive pressures may be developed, causing intense pounding.



CLEARANCE MEASUREMENTS

Distance from Top. Inches	Cylinder Measurements		Piston Measurements	
	A	B	A	B
3	16.025	16.020	15.975	15.980
10	16.030	16.029	15.980	15.982
18	16.029	16.019	15.975	15.982

FIG. 116.—Log of piston and liner wear.

In a discussion before the English Diesel Engine Users Association it was stated that piston and liner wear depends upon the following factors:

Ash.—Tar, creosote oils, and petroleum fuels, oils having an asphaltic basis, form ash some of which may be, and generally is, of a highly abrasive character. The permissible ash content should not exceed 0.08 per cent, of which proportion not more than one-half should be a gritty nature.

Quality and Quantity of Lubricant.—The general consensus of opinion tends to the use of a pure mineral oil. This matter was very thoroughly ventilated in a Paper read to the Association by E. A. Evans, on January 26, 1915, and in the subsequent discussion. The quantity used must be determined for each engine. An excessive quantity is deleterious, as it will burn and form carbon deposits on the piston, piston rings and liner, and assist liner wear by restricting the freedom of the piston rings.

Method of Lubrication.—Most modern engines are fitted with pressure lubrication for pistons and liners. Splash lubrication is being abandoned. Many engineers object to splash lubrication as being, in their opinion, an inefficient method. One of the engines under the care of a member is arranged with splash lubrication, and has never given rise to any anxiety. It is necessary, of course, to examine the lubricating holes below the piston head occasionally to ensure they are all clear. One very bad seizure has occurred from neglect of this precaution.

The Fit of Pistons and Piston Rings in the Liner.—Uncooled pistons up to 18-in. diameter, should be fitted about 0.008-in. clear of the liner at the top (on a diameter); about 0.010 in. at the ring grooves; about 0.010 in. at the gudgeon pin holes; about 0.008 in. at the skirt. These clearances should be sufficient to allow for growth. Pistons above 18 in. should be given an additional 0.002 in. in each position. Water and oil cooled pistons may be given rather finer clearances. Piston-ring pressure is estimated to range from 5 to 6 lb. per square inch, and it is desirable to chamfer the rings, removing the sharp edges. Concentric rings are now almost universal. General experience shows that a larger number of thin rings are more effective than a smaller number of wide ones. Wiper rings are sometimes fitted below the gudgeon pin. The major part of liner wear is caused by the first three rings from the top of the piston.

Side Thrust Due to the Connecting Rod.—Side thrust is influenced by the length of the connecting rod and the ratio between the length of the rod and that of the crank. The wear from this cause should be in direct proportion to the bearing area (projected) of the piston in the liner. Table XXII, gives particulars, including side thrust, of several engines. The thrust values from indicator diagrams taken from the several engines have been calculated. These values are not strictly comparable as the engines were not all loaded to the same fraction of their normal rated outputs when the diagrams were taken. The crank angles, at which the various maximum side thrusts occur, merit a little consideration. For maximum side thrust with a constant pressure expansion line, the connecting rod and crank should be at a right angle to each other, *i.e.*, the crank angle should be about 78 deg. Owing to the rapid drop of pressure in a Diesel engine during the expansion stroke, the maximum effect of the connecting rod occurs a good deal earlier in the stroke. The table shows calculated side pressures from 30 to 40 lb. per square inch. The diagrams from which the particulars were obtained were taken under ordinary work-a-day conditions, and are not put forward as examples of the best practice under test conditions. It is desirable to ascertain if possible to what extent liner wear and side thrust are interconnected, but, as the figures of wear are conflicting, no definite decision has been made. The values of side thrust being all very similar, the point is not of much importance for comparative purposes, so far as these notes are concerned. Other factors are of greater importance.

TABLE XXII

Engine	Cylinder diameter, in	Ratio stroke bore	Ratio rod crank	Piston length, in	Mean effective pressure per sq in	Fraction of rated maximum load, per cent	Piston speed (ft per min)	Side thrust	
								Lb per sq in	Crank angle, deg
g 1	12	1.52:1	4.1:1	27¼	93.0	100.0	760	37	35
g 2	12	1.52:1	4.1:1	27¼	96.8	90.0	760	34	37
g 3	20	1.2:1	4.5:1		87.0	100.0	800	33	25
4	13	1.52:1	3.9:1	30	80.4	82.0	800	31	43
h 5	12	1.507:1	5.31:1	26⅛	.	83.6	755	31	39
6	17¾	1.47:1	5.27:1	37½	90.3	78.3	758	20	26

Quality of Material.—No attempt to discuss this very important matter is made. It is one for a metallurgist to deal with, as it involves such difficult and highly-technical considerations as the composition and structure of cast iron, growth of cast iron, etc.

Piston Ring Troubles.—Beyond an occasional broken ring the only difficulty that the operator will experience is the gumming of the rings in the grooves. The gumming may be produced either by an excessive amount of fuel oil which remains in the cylinder in an unconsumed condition or by an overabundance of lubricating oil. The solution of the latter trouble is simple; all that need be done is the reduction of the quantity supplied. If the oiling is accomplished by a mechanical pump, the control can be readjusted. If the engine has splash lubrication, the oil level can be lowered, thereby preventing undue throwing of the oil emulsion on to the cylinder walls. Where the trouble is traceable to excessive fuel oil, the problem is not so simple of solution. It is apparent that, in such cases, the difficulty lies in the fuel valve. The timing may be incorrect, or the air pressure too low or too high, or the atomizer disks or cones may be out of order. It becomes necessary to try various experiments until the proper remedy is obtained.

The rings can be loosened by soaking the entire piston in kerosene or lye water, after which the rings can be pried off. In removing the gum and carbon a copper or brass scraper should be used. It is not advisable to clean with emery paper as long as the piston is perfectly smooth. Each time the engine is laid up for a few hours a small quantity of kerosene should be injected into the cylinder through the air admission valve. This kerosene will remove any carbon on the piston and rings that is in the process of formation. The exhaust valve, in such event, is best blocked open to allow the vapor to escape.

Grinding Taper Piston Pins. A number of engines have piston pins with taper ends. The tapered ends fit into ground seats in the piston bosses. The ground seats gradually pound out of round, forcing the engineer to regrind the pin to new bearings. To accomplish this with the minimum of trouble the piston can be placed on two 8 by 8 in. timbers, as shown in Fig. 117. A discarded valve spring, if set under the piston immediately below the pin, will keep the pin raised from the seats in the bosses unless the pin is pressed downward. Emery paste is coated over the two pin ends, and while the pin is forced into the bosses it is rotated by a pin wrench, as outlined in the sketch. Removal of the downward pressure allows the spring to raise the pin. This action serves to distribute the grinding paste over the entire seat, preventing the seat from being ground hollow. The pin grinding is fully as important as valve grinding although it is done very carelessly in many plants.

In reassembling the piston, pin and rod it is not enough simply to push the pin into the bosses and tighten up on the lock-nut. The pin must be driven in with a sledge and copper mallet with as great a blow as an able-bodied man can deliver. The timorous feeling many engineers have about this driving process is entirely without foundation; there is no danger of fracturing the piston.

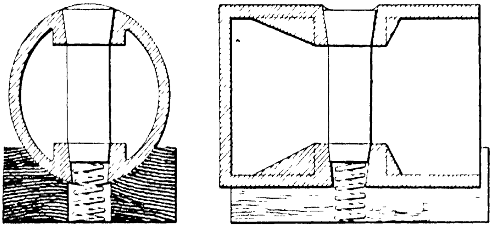


FIG. 117.—Grinding piston pin in taper bosses.

Emergency Piston Pin.—When a pin has scored badly or developed a fracture, it occasionally becomes necessary to make an emergency pin. Without question the manufacturer is the proper party to furnish this new part. But where the engine is urgently required, the engineer cannot wait on the slow delivery that is so prevalent. A makeshift pin can be turned from cold rolled shafting. It should be machined to size and then case-hardened with bone-black or cyanide of potassium. After remaining in the fire for twelve hours, the pin should be cooled off. As a rule, the pin expands but slightly in the hardening process. The exact dimensions can be secured by lapping with emery paste, although it is a laborious procedure.

CHAPTER X

CONNECTING RODS

Types and Adjustments

American Diesel Engine Co.'s Connecting Rod.—The first Diesel manufactured in America was fitted with a connecting rod similar to Fig. 118. This gave place to the type illustrated in Fig. 119. The wedge design was quite prevalent in gas engine work, and the Diesel builders evidently obtained it from that source.

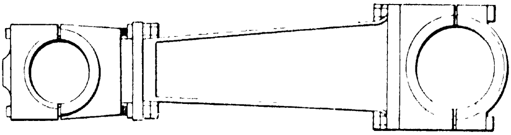


Fig. 118.—American Diesel Co. connecting-rod.

In adjusting the piston-pin bearing, it is necessary to remove the rod from the piston. To reduce the clearance resulting from the wearing of the brass, the cap is unbolted and the required amount of shims removed. The wedge must then be brought up snug against the brass. Some operators, under a mistaken idea, use no shims or separators, depending solely upon the wedge. It must be conceded that many engines have operated fairly successfully under this condition. However, the shims serve to keep the two bearing halves rigid, and, in the

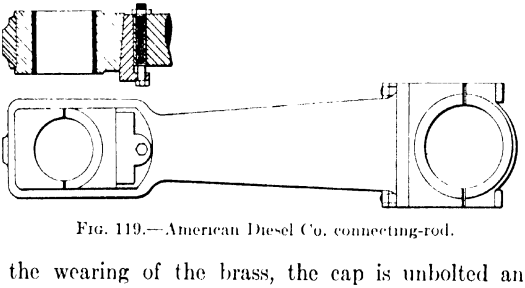


Fig. 119.—American Diesel Co. connecting-rod.

event wear occurs, the play of the two halves will not hammer the wedge. The objection to this form of wedge is based on the frequency of fracture of the wedge bolt. The wedge has a steep angle, and the end thrust against the bolt is of considerable proportion. The break usually occurs in the thread, allowing the wedge to shift. A remedy for this can be obtained by reducing the wedge angle. This, of course, lessens the total adjustment obtainable, but shims can be interposed between the wedge and brass, giving a further lift. Such breakages are always on engines where the piston-pin bearing clearance is allowed to be excessive. The hammer blow given the bearing when the crank passes top center and the rod starts downward will eventually mash the bearing or break the wedge bolt.

The crank or big end follows conventional Diesel lines in being of the marine design. The housing is cast steel and has the babbit run directly on its inner surface.

Snow Vertical Diesel Rod.—The vertical Diesel built at the Snow Works has a rod with a marine big end and a bored piston-pin end, as shown in Figs. 75 and 111. The big end is unusual in that four bolts are used in place of two as is customary.

The upper end carries a two-piece white-metal lined brass. Adjustment for wear is obtained by removing shims and drawing down with two adjusting screws at the top. As with all other rods the pin must be removed from the piston when the pin clearance is to be altered, fitted to the bearing and then replaced in the piston.

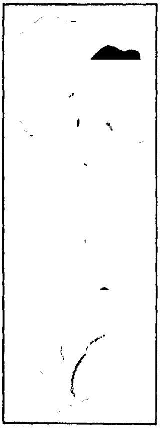


FIG. 119A.—Rod of McI. & S. Diesel.

McIntosh & Seymour Diesel Rod.—The trunk piston engine employs a rod similar to Fig. 122. The crosshead engines use a rod, Fig. 119A, with a forked upper end and a marine lower or crank-pin end. The crosshead carries a slipper-type shoe which bears on guides bolted to the frame standards.

Allis-Chalmers Diesel Connecting Rod.—An improved form of the wedge-type rod is found on the Allis-Chalmers Diesel. This rod has a marine crank end and a wedge-adjustable piston-pin bearing, Fig. 120. The improvement consists, mainly, in the employment of adjusting or separating set-screws in conjunction

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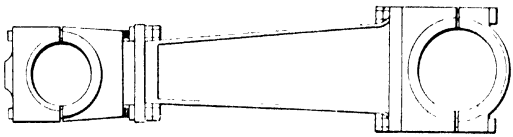


Fig. 118.—American Diesel Co. connecting-rod.

In adjusting the piston-pin bearing, it is necessary to remove the rod from the piston. To reduce the clearance resulting from the wearing of the brass, the cap is unbolted and the required amount of shims removed. The wedge must then be brought up snug against the brass. Some operators, under a mistaken idea, use no shims or separators, depending solely upon the wedge. It must be conceded that many engines have operated fairly successfully under this condition. However, the shims serve to keep the two bearing halves rigid, and, in the

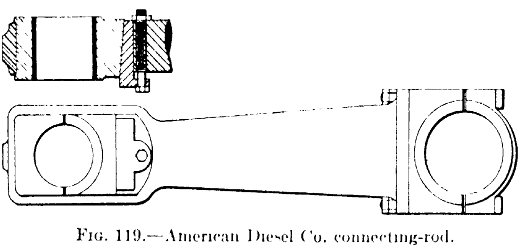


Fig. 119.—American Diesel Co. connecting-rod.

Adjustment at the piston-pin end reduces the connecting-rod length, and thereby increases the compression clearance in the cylinder.

In compensating for the wear at the big end, the wedge is taken up. Since the entire pressure acts on the wedge, the bolts must be kept snug; also there must be no binding of the bolt if fracture at the thread is to be avoided.

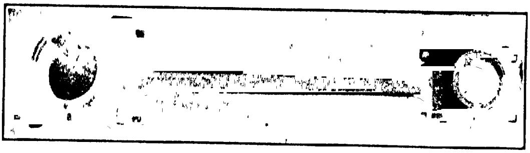


FIG. 122.—McEwen Diesel connecting-rod.

National Transit Diesel Connecting Rod.—The National Transit Pump & Machinery Co. has adopted the form of rod appearing in Fig. 123. The big end is of the standard marine type while the piston-pin end is solid and fitted with bronze bearing shells. The wear on these shells is compensated by the adjusting screw in the rod end.

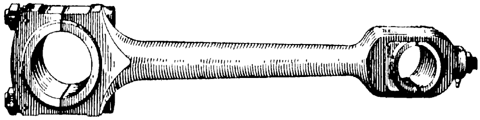


FIG. 123.—National Transit Diesel connecting-rod.

With this particular rod, since the big end does not admit of the use of separators to bring the rod length back to normal, this correction must be made by the insertion of strips or shims between the piston-pin lower bearing shell and the rod.

Busch-Sulzer Bros. Diesel Connecting Rod.—The rod design of the four-cycle engine has the big end of the marine type, separate from the rod itself. The piston-pin end is solid, having phosphor-bronze bearing shells and an adjusting screw, as outlined in Fig. 124.

Since the big end is separate, the rod length is corrected by the insertion of separators between the rod and the big end. The crank pin is lubricated by a drilled passage in the shaft, as indicated in Fig. 80. The rod is drilled its entire length and the piston pin receives its lubrication through this passage. The passage in the rod registers with the oil line in the crankshaft once per revolution, thus obtaining the proper amount of lubricant.

De La Vergne Diesel Connecting Rod.—On the F.D. Diesel engine manufactured by the De La Vergne Machine Co. the rod follows the lines of Fig. 125. This is quite like the two rods last mentioned. The unique feature is the drilled passage in the center of the rod for the purpose of oiling the piston pin.

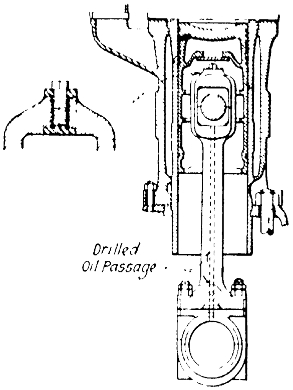


FIG. 124.—Busch-Sulzer Diesel connecting-rod.

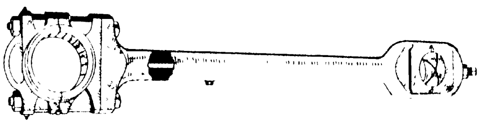


FIG. 125 —De La Vergne connecting-rod.

The oil is picked up from the big-end bearing, and the ports here must be kept open. It is possible for a hot big-end bearing to close the ports with flowing babbitt. This, of course, results in a ruined piston-pin bearing.

New London Ship & Engine Co. Connecting Rod.—The Nelseco Vertical Marine Diesel employs a connecting rod with a marine big end and a solid piston-pin end. The latter end has a bronze bushing but in the small sizes is provided with no means of adjustment. If the bushing wears, it must be replaced with a new one; in an emergency the bushing can be reduced at the split and a thin shim interposed between the bushing and the walls of the rod end. The piston pin is lubricated through a drilled passage in the rod.

Fulton Diesel Connecting Rod.—The connecting rod of the Fulton Diesel is illustrated in Fig. 126. The bearing for the crankpin is somewhat unusual. The bolts are not a drive-fit, as in many engines but, instead, the holes are drilled to allow radial clearance. The two halves of the housing form a male and female joint. The result is that there is no bending stress in the bolts. This, the builder claims, largely removes the trouble of broken big-end bolts.

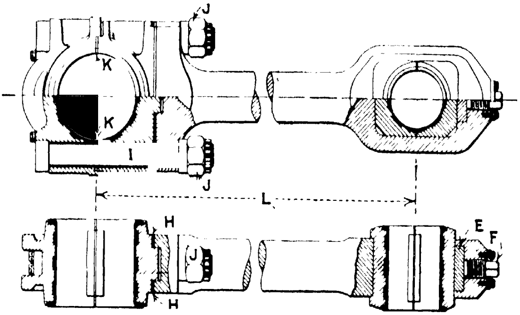


Fig. 126.—Connecting-rod of Fulton engine.

The piston bearing is adjusted by the top screw *F*. This, after adjustment, is held fast by the locking plate.

Standard Connecting Rod.—A rod out of the ordinary is that on the Standard Diesel. The piston-pin end is in the form of an eye. The bearing halves fit into the eye; adjustment for clearance is obtained by removing shims or filing the bearing edges and tightening up with the top adjusting screw.

Winton Diesel Rod.—The Winton Diesel shows in the rod, Fig. 71, the characteristics of the factory as the piston end of the rod is quite like that used on some automobiles; the Winton factory being an outgrowth of the Winton automobile factory. The pin is not fastened in the piston with a bearing in the rod as usual. The pin is provided with two end bearings pressed into the piston while the rod is clamped to the pin. There is then but slight opportunity for adjusting the wear of the bushings. The big end follows the standard practice, although four bolts are used in place of the usual two. Lubrication of the piston pin bearing is through a small pipe leading from the crank bearings up to the hollow piston pin.

Dow Connecting Rod.—The Dow Diesel employs a standard marine-type rod.

Lombard Connecting Rod.—The Lombard connecting rod, see Fig. 71 *A*, is provided with a marine-type crank end and an eye piston-pin end. The piston-pin bearing is a two-piece shell and is not provided with means of adjustment.

Lubrication of the piston pin is accomplished by a pipe attached along the rod and which receives oil from the crank-pin passage and delivers it through a drilled passage in the piston pin.

Nordberg Connecting Rod.—The Nordberg engines are all two-cycle with crossheads. This requires a modification of the usual rod by splitting the upper end into two parts. Each part carries a bearing into which the ends of the crosshead or crosshead pin fits. The piston rod is screwed into the crosshead block as shown in Fig. 127. The crosshead block carries two shoes of the bored type.

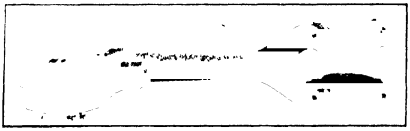


FIG. 127.—Nordberg connecting-rod.

Busch-Sulzer Type C Connecting Rod.—The connecting rod of this engine is of the marine type quite similar to the rod of the Type B engine. It is shown in Fig. 82.

Big-end Bearings.—No matter what the design may be, the operator some day is confronted with the problem of a big end that insists on running hot. The first move is to determine whether the lubrication has been faulty. In the majority of cases this proves the origin of the trouble. Generally the oil pipe or passage has become clogged with dirt or a bit of waste. The remedy is obvious. There is, for some unknown reason, a tendency for the big-end bearing to wear more rapidly at one end than at the other; or, at times, both ends wear while the center remains in its original condition. This "belling" of the bearing permits the pin pressure to be distributed over a rather small area of the brass. This produces a local heating that forces the babbitt to drag, filling the oil passages and grooves. An additional result of this unequal bearing wear is the scoring of the cylinder on one side. When this bearing wear has occurred, it is imperative that the babbitt be rebored to the pin diameter and the oil grooves cut. Even if the wear seems excessive, it is, as a rule, possible to avoid rebabbitting. Part of the shims between the bearing halves can be removed, and the halves clamped together and rebored to size.

It sometimes happens that part of the babbitt cracks and drags around the pin. This results in heating and a badly scored bearing. If the trouble is local, the rough spots can be smoothed with a scraper, and the bearing can then be placed in service again. In all instances where big-end bearings become so hot that the babbitt is thrown, the engine should not be stopped immediately; rather, the load should be taken off and the engine turned over very slowly, with the particular cylinder cut out. In rebabbitting rod bearings the same method as described for main bearings can be followed. The babbitt should always be cut on a bevel at the junction of the two halves. The oil grooves should not extend to the bearing edges, and, when a pressure-oiling system is used, the oil grooves should be eliminated since they allow the oil to escape too rapidly.

Side-play is of frequent occurrence in Diesel operation. The best method is to thin the bearing sides and run a collar of babbitt around the bore. In addition the alignment of the rod bearing should be corrected. This collar must be turned square with the pin. It is not necessary to cover the entire side; consequently, the collar can be machined parallel with the side of the big end. Another method is the employment of sheet-steel washers between the web and the big-end sides. Inspection of the connecting-rod bearings should be performed at least every three months.

Along with the wear of the big-end bearing occurs the wear of the pin. In old engines, which have seen several years of service, the pins may become flattened on one side. This can be corrected by filing and lapping, but it is a task requiring great care and patience. It is, ordinarily, not difficult to detect a worn rod bearing. The engine will emit a thump or pound both on the in and out strokes.

Rebabbitting Bearings.—The engineer will some day face the proposition of renewal of the pin bearings. In many engines the piston-pin or crosshead-pin bearings are phosphor-bronze or Parsons white-metal shells without babbitt linings. These bearings may become scored or pounded out of shape. It is only necessary to place the box on a lathe and rebore to the correct diameter. If there are not enough shims by removal of which the two bearing halves may be brought together close enough to allow reboring, the edges may be faced off on a planer or lathe, followed by reborings.

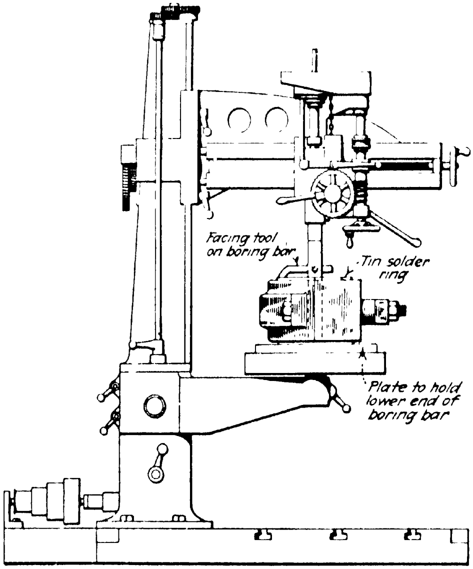

Facing tool on boring bar
Tin solder ring
Plate to hold lower end of boring bar

Fig 128.—Boring crankpin bearing.

It is in the rebabbitting of the crankpin bearing that the engineer will find some difficulty. The process is the same as for mainshaft bearings. In the boring after babbitting it is essential that the big-end bearing be bored exactly square with the side faces. These faces are always made square with the big-end flat surface when machined at the factory.

To machine the bearing, the housing should be placed on a lathe carriage and the boring tool held on the face plate, or better still, bored on a drill press if a good one is at hand. A plate with a center hole to hold the end of the boring bar should be bolted to the drill press table. The housing should be leveled up on the table, and with the cutting tool in the boring bar the spindle should be turned by hand at first. In this way it is possible to see that the bar is central with the housing and that the housing is square with the bar. The method is outlined in Fig. 128. Such a method will give a bearing bore that is exactly square with the faces and with the flat surface upon which rests the big end of the rod.

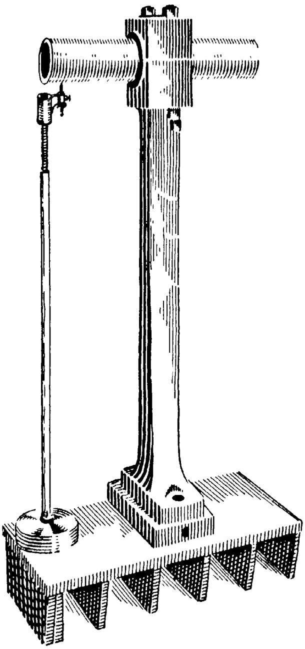


FIG. 129.—Connecting-rod gage.

Checking Rod Alignment.—After rebabbitting and boring the assembled rod should be checked for alignment. If the bores of the piston-pin and the crankpin bearings are not parallel, the connecting rod will not move in a vertical plane but will tend to bell the crank bearings as well as to pound badly. Many instances of lengthwise motion of the shaft are due to error in one or more rods.

To test the rod it is necessary to obtain two mandrils of the same diameters as the two pins. These may be turned up out of cast iron and can be made by any mechanic who can handle a lathe. If pipe of the correct diameter is handy, a piece machined true makes a light serviceable mandril. There is no reason why the mandrils may not be used as the mandrils for babbitting as well.

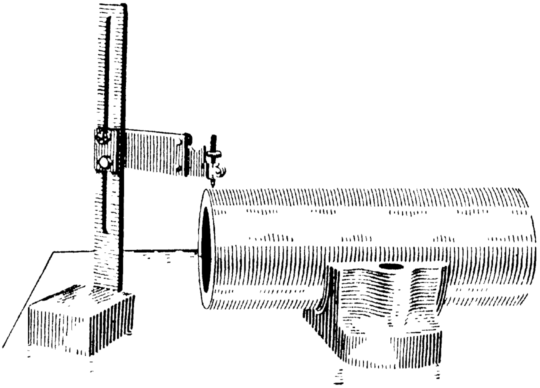


FIG. 130.—Gaging big-end bearing alignment.

A gage is made up consisting of a brass disc (steel will do as well) to which is attached a threaded rod of ¾-in. drill steel. The top of the rod is threaded to receive a brass bushing carrying a sharp point, as shown in Fig. 129. The piston-pin mandril is put in the bearing and the bearing halves clamped tight. The rod is now placed on a flat surface, such as the table of a shaper or similar machine. In fact, every engine plant should have a surface plate. The gage is adjusted until the tip will just touch the mandril as shown. By passing the gage under the opposite end of the pin it is at once apparent if the mandril is square with the big-end surface of the rod.

To check the crankpin bearing the top half of the bearing is laid flat on a surface (Fig. 130). If it is necessary in order to clear the center boss, the bearing may be supported on thin blocks and leveled with a spirit level. By using a surface gage across the tops of the mandril ends the correctness of the bearing is quickly determined.

It is possible that the pins are square with the rod and yet may not lie in the same plane. To check this the rod is placed as shown in Fig. 131 with one mandril supported on V blocks and the other end held up by packing pieces. The surface gage will at once determine the fact that the pieces are true or out of parallel. This procedure may seem tiresome to some but Diesel engineers must understand that care is necessary and with such precautions a quiet sweet-running engine will result.

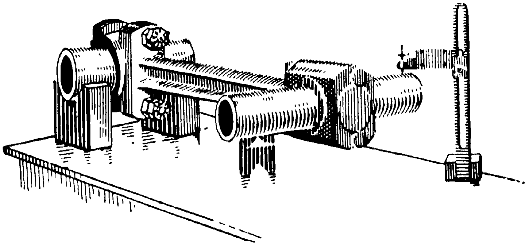


Fig. 131.—Checking parallelism of the pins.

After a bearing is rebored, it should be scraped in the same manner as a main bearing.

Pin Clearance.—To estimate the amount of clearance necessary between bearing and pin, an excellent scheme is to "jump" the rod with a bar. If a slight movement can be felt, the big end has ample clearance. "Jumping" the piston allows one to judge the piston-pin clearance. Another and better method is to tighten up the piston pin bearing until the connecting-rod can be barely swung back and forth when the piston is suspended by a chain hoist. If the movement is free, there is too much clearance. If a man cannot swing the rod without tilting or moving the piston, the bearing is too tight. This method is shown in Fig. 132. A clearance of 0.0003 to 0.0005 in. per inch of pin diameter is about right.

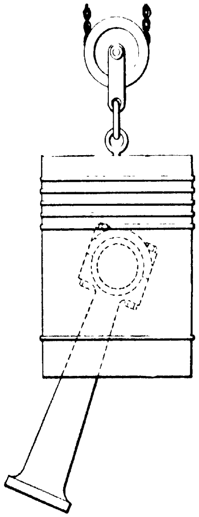


Fig. 132.—Swinging rod to check clearance.

Removing Rod.—If the rod is to be removed, in most designs the piston and rod must be pulled through the cylinder after removing the cylinder head. However, if the crankpin bearing only is to be removed for inspection or measurement of clearance, the piston need not be taken out. In trunk piston engines the piston can be wedged in place by a timber *b* resting on a cross timber and supporting the bottom edge of the piston, as illustrated in Fig. 133. The big-end bolts are now removed and the bearing halves lifted out. To measure the clearance leads are placed on the bottom half of the bearing and the two halves are drawn together by bolts, using a collar or larger nut in place of the bottom end of the rod.

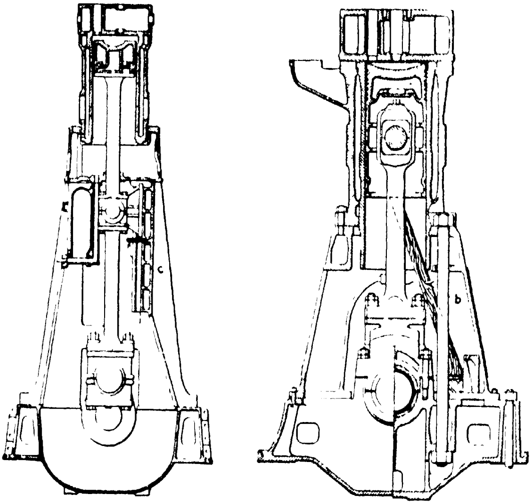


FIG. 133.—Methods of holding piston.

On large crosshead-type engines the bearings are too heavy to handle without block and falls. The crosshead may be held up by tapping a hole, in one guide and inserting a pin as shown at *c* in Fig. 133. By using two small blocks and falls, with eye bolts screwed into tapped holes in the bolt ends and small enough to pass through the nuts, the lower bearing half may be lowered with ease, as illustrated in Fig. 134.

In many engines the valve cages may be removed and eye bolts screwed into the piston head and the latter held by small blocks and falls.

Before removing the nuts a series of graduations should be marked off and numbered and an arrow placed on bolt end.

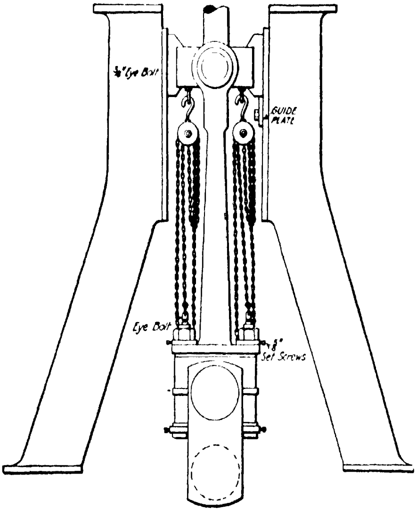


FIG. 134.—Lowering bearing with small blocks and falls.

Bearings of Crosshead Type Engines.—The construction of the crosshead in most large Diesel engines follows conventional marine steam-engine practice, the two crosshead pins projecting from the forged crosshead block and the upper end of the connecting rod being forked to support two sets of bearing boxes. In most existing engines the space around the crosshead is restricted to such an extent that it is not possible to use chain falls or other lifting gear when adjusting; but this does not act to handicap the work, for in most cases the crosshead-pin boxes are small enough to be readily handled by hand.

The crosshead being worked on is jacked down to its lowest position and leads inserted to measure the pin clearance. The leads, instead of being laid in the lower box, as in the case with the crankpin bearing, are laid across the top of the crosshead pin, the reason for this being that the weight of all the moving parts above the connecting rod is supported by the lower boxes, and when the engine is stopped the clearance is all on top of the pin. After the leads are taken and before any adjustments are made, the babbitt lining in the lower box should be examined for signs of uneven wear, wiping, cutting, etc. This can be done by hanging the crosshead slipper on the stop bar through the guides and jacking the crankshaft around until the connecting rod moves away from the crosshead pins far enough to expose the bearing surface of the lower boxes.

If the engine has been in service long enough to warrant expectation of appreciable wear of crosshead pins, the pins should be examined for shoulders. Unlike the crankpins and main bearings, the crosshead pins do not have a rotative motion within the bearings. As the lower end of the connecting rod rotates in a circle the radius of which is the crank radius, the upper end oscillates so that the boxes turn on the crosshead pins in one direction during half a revolution of the crank and in the opposite direction during the remainder of the revolution. This rubbing of the boxes back and forth on the pins will sometimes cause a shoulder to wear on each side of the pins, and it is necessary to file this shoulder down before the bearing can be properly adjusted. A very good way to prevent the occurrence of these shoulders is to cut away the pin so that there is a flat spot on each side. This allows the bearing surface of the boxes to override the edge of the cut each time, and no shoulders can form. This does not reduce the effective bearing surface of the pins, because this effective surface is only on the top and bottom portions of the pin.

Big-end Bolts.—A source of trouble in some types of Diesels has been the big end. This is in main due to poor bolt material and to excessive sledging-up of the bolt.

Various materials have been used, such as, wrought iron, mild and high carbon steel, the latter being quite generally used at present. The specifications below are those of a prominent Diesel builder, and if the operator decides to have bolts made rather than to purchase from the manufacturer, these requirements should be followed.

SPECIFICATION

Carbon: 0.30 to 0.40 per cent. Silicon: 0.30 max. per cent
Manganese: 0.50 to 0.80 per cent. Nickel: 2.75 to 3.25 per cent.
Sulphur and phosphorus per cent. as low as possible.

The material to be oil hardened and tempered to give the following values:

Tensile strength: Not less than 50 tons per square inch.
Yield point: 65 per cent.
Elongation: 20 per cent.
Reduction of area: 45 per cent.
Impact value: 40 ft.-lb.

The ordinary life of a bolt varies from 2,000 hr. to 30,000 hr. of service. Probably 20,000 represents the period of service advisable before replacement. Each time a bearing is removed the bolts should be inspected. This consists of polishing the bolt and then coating it with oil. The oil is then rubbed off and a coat of white lead is rubbed on. If there are any minute incipient cracks they will show up by the oil oozing out, making a mark on the white lead. In addition it is a good plan to go over the bolt with a strong magnifying glass.

The operator is cautioned against running with excessive clearance at the pin. With a large clearance there is a hammer blow resulting from the inertia of the parts as dead center is reached. This shock must be borne by the bolts over and above the initial stress of sledging up.

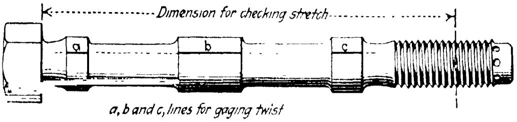

Dimension for checking stretch
a
b
c
a, b and c, lines for gaging twist

FIG. 135.—Checking elongation of rod bolts.

In sledging up the nuts it is by no means unusual to find that an excessive initial stress has been placed on the bolts. With an added stress from inertia forces the bolt may break. The engineer should see that the bolt is tightened up snug by a hand wrench. A sledge may then be used to give the nut a final tightening, but under no circumstances should the sledging be continued for more than a few blows.

To detect elongation of the bolts one builder files a flat on one of the threads, placing a center-punch hole here as well as one just under the head. A line is scratched along the shank in line with these prick marks, as shown in Fig. 135. By keeping the original distance between the marks in a data book, it is possible to discover any elongation upon inspection. Likewise the marks on the shank will show if the bolts are twisted.

Still another method is illustrated in Fig. 136, where a gage fits into a dowel hole in the head and has an adjustable screw which rests on the bolt end as shown. By filing each bolt to a uniform length it is possible to have a single gage for all cranks. It is easy to check the bolts while on the rod. This is a most attractive gage and is easily made.

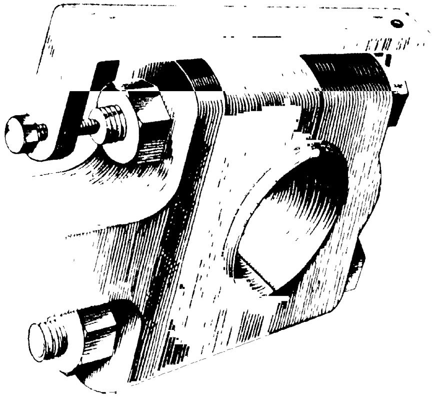


FIG. 136 Second method of checking bolts.

Annealing of Bolts.—Although some experts recommend annealing of the rod bolts after a year's service this is not good practice. Annealing will help an over-stressed bolt but little. New bolts each year or eighteen months is a better plan.

CHAPTER XI

MAIN BEARINGS, CRANKSHAFTS

TYPES, ADJUSTMENTS, AND REPAIRS

Main Bearings.—Diesel engine bearings may be divided broadly into two classes; namely, adjustable and non-adjustable. The former, of course, embodies the addition of a wedge and, as applied to the vertical type engine, is outlined in Figs. 13 and 27.

Adjustable Bearings for Vertical Engines.—This bearing deisgn is found only on the older engines, such as the American Diesel. It follows gas-engine practice of fifteen years ago. At that time the rapid wear of the main bearings seemed to demand some form of adjustment. It is now apparent that the trouble lay in excessive bearing pressures and non-homogeneous babbitt and very flexible shafts. With more liberal bearing area this difficulty of rapid wear has largely disappeared.

If the engine is in the hands of an experienced and skilled operator, the adjustable bearing is not dangerous. On the other hand a less intelligent mechanic will quite likely damage the engine when adjusting the bearing. It is well-nigh impossible to have three or four bearings with babbitt liners of uniform characteristics. The babbitt first run out of the ladle is of a density considerably higher than the bearing that is poured last. The natural result of this non-uniform density is the variation in the bearing wear. To the engineer who is versed in the finer adjustments of a Diesel engine the realignment of the bearings is not a difficult task.

Adjustable Bearings on Horizontal Engines.—The direction of the bearing pressure, due to the explosion in the cylinder, makes the adjustable bearing well-nigh imperative on horizontal engines. The cylinder pressure against the piston head may be said to act in three directions. Part acts in a downward direction against the cylinder walls, the remainder has its direction along the connecting rod. This latter force is separated into two components, one acting tangentially to the crankpin, producing

Still another method is illustrated in Fig. 136, where a gage fits into a dowel hole in the head and has an adjustable screw which rests on the bolt end as shown. By filing each bolt to a uniform length it is possible to have a single gage for all cranks. It is easy to check the bolts while on the rod. This is a most attractive gage and is easily made.

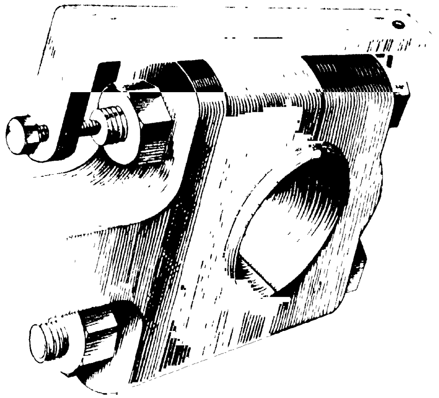


FIG. 136 Second method of checking bolts.

Annealing of Bolts.—Although some experts recommend annealing of the rod bolts after a year's service this is not good practice. Annealing will help an over-stressed bolt but little. New bolts each year or eighteen months is a better plan.

cant since the line of pressure, Fig. 137 never falls along the bottom when the cylinder pressure is high; consequently the lower shell receives but little pressure other than that due to the weight of the flywheel and shaft. The front quarter experiences the greatest wear and must be given constant attention. Ordinarily it will be found that the two main bearings do not wear at a uniform rate. While the wear on each bearing should be taken up as it develops, it should not be forgotten that the lower shell and the rear quarter also require attention. Periodically the shaft should be realigned and, if it is proven out of line, the lower and rear shells shimmed or wedged up the proper amount. Since the clearance space between the cylinder head and the piston is small in the Diesel engine, there is a danger of excessive compression pressures if the wear on both front and back quarters is taken up by the front quarter only, for this tends to throw the shaft toward the cylinder. Usually the horizontal engine employs an extension shaft and outboard bearing. This outer bearing along with the main bearing adjacent to the flywheel carries the weight of the wheel. The flywheel weight on this main bearing causes more rapid wear on the bottom quarter than on the other main bearing. Realizing this the operator should see that the bearing does not become low.

The engineer will find that the bottom bearing wears most at the edge adjacent to the front quarter. The front quarter also wears at this edge.

It should be a rigid rule in every plant where oil engines are installed that no adjustment of the bearings shall be made while the engine is running. The revolving shaft allows the bearings to be drawn up tighter than when idle; the careless operator will likely tighten up the wedges enough to occasion a hot bearing.

Adjustments.—In readjusting the bearings the first step is to remove the bearing tops, then thoroughly wipe off the shaft and bearings after the shaft has been raised by a jack set at each end of the frame. After the shaft is lowered on to the bottom bearings, it should be aligned by using a spirit level. The two end bearings are adjusted until the shaft is level. The inside bearings are then brought up snug against the shaft by means of the wedges. The connecting rod big-ends are next made fast and the clearance of each cylinder measured. Often, when the engine has been badly out of alignment, the readjustment of the bearings alters the clearance of one of the end cylinders. This, of course, must be taken care of by varying the thickness of the shims between the connecting rod and its big-end.

The operator must be cautioned in one particular. If the engine shaft carries an overhung flywheel or if the outboard bearing beyond the flywheel be low, the shaft will bow in the center. When the center bearings are brought up against the shaft they are actually out of line and hold the shaft in its bowed position even though the outboard bearing be realigned. The result may be a broken shaft. A shaft bent in this fashion is subject to flexure or bending twice per revolution. Any shaft has a definite number of flexures, depending upon the size, which it will undergo before breaking. A small addition to this critical number will invariably cause the shaft to fail. It follows that an overhung wheel must be light weight or the shaft must be larger than necessary for strength if danger of fracture is to be avoided.

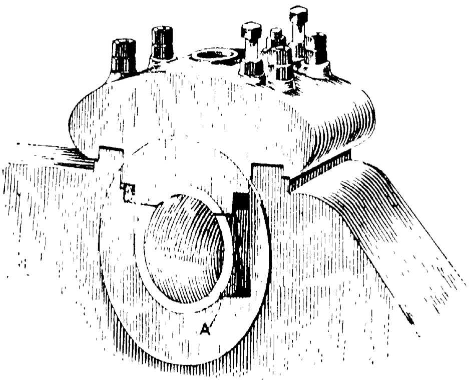


FIG. 138 – National Transit main bearing.

National Transit Diesel Main Bearing.—Figure 138 is the three-piece bearing used on the National Transit Diesel. As will be noted, the upper part of the bearing is babbitted directly on to the bearing cap. The bottom and rear quarter consists of a single babbitted shell while the front quarter is provided with a wedge. To avoid the danger of wedge-bolt breakage, the bolt-head fits into a circular opening in the wedge. The bolt can, then, accommodate itself to any displacement of the wedge.

The lubrication is effected by two chain oilers which dip into the oil cellar cast in the housing.

The Snow Oil Engine Main Bearing.—This horizontal engine employs a bearing as illustrated in Fig. 139. The lower shell, as well as the two quarter-boxes, is equipped with a wedge. The top quarter is a separate shell and is not cast directly on to the cap, as is usual. The oiling of the bearing is accomplished by means of a mechanical lubricator furnishing stream lubrication.

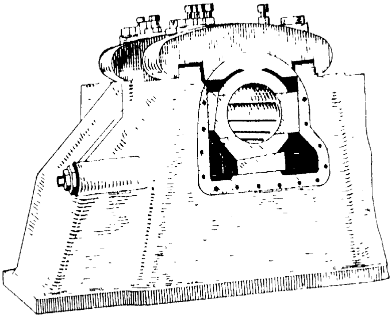


Fig 139.—Snow main bearing.

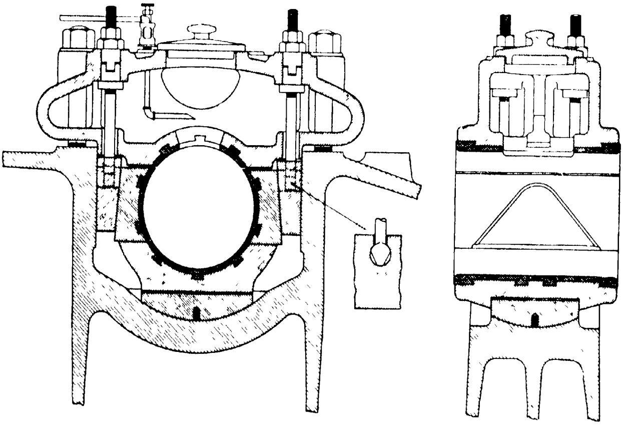


Fig. 140.—Allis-Chalmers main bearing.

Allis-Chalmers Diesel Main Bearing.—A four-piece bearing has been adopted by the Allis-Chalmers Co. and appears in

Fig. 140. The lower shell rests on a spherical-bottomed pad with the idea of allowing the bearing to shift to conform with the position of the shaft. Since the shaft has but one true position, which must be maintained if the operation be smooth, the spherical seat is not absolutely necessary. In shimming up the lower shell, the liners are properly placed between the shell and pad. The two side quarters are equipped with wedges, while the top is babbitted directly onto the cap.

The operator, especially if he has had previous experience on steam engines, should remember that the adjustment must be made almost entirely by the front wedge and not by both quarter-boxes as in steam practice. The wedge bolts are of a design that eliminates any danger of breakage. The oiling is secured by stream lubrication from a mechanical oil pump.

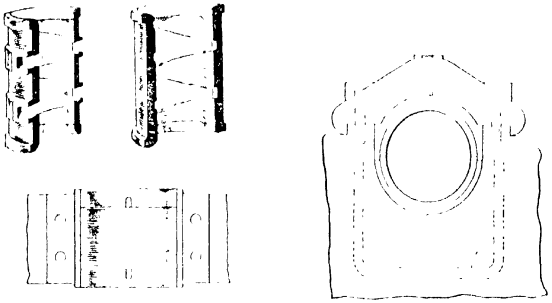


Fig. 141 McEwen bearing

McEwen Diesel Main Bearing.—Figure 141 shows the two-piece bearing of the McEwen Horizontal Engine. Since these engines are small-powered per cylinder, the two-piece construction offers no objection. It should be understood by the operator that the shaft pressure will wear the bearing oblong and that no adjustment is practical beyond the removal of the shims. After the wear becomes pronounced, the only remedy is replacement of the worn part. The oiling is effected by two rings which dip into the oil cellar below the bearing.

De La Vergne Diesel Main Bearing.—The De La Vergne engines use a two-piece bearing, as appears in Fig. 142. The bottom and top halves are set at a 45-deg. angle which allows the pressure to bear on the center of the bottom shell. By removing shims from between the two shells the necessary adjustment can be made.

Otto Diesel Main Bearing.—The Otto Diesel has the two-piece bearing, adjustment of which is secured by shims.

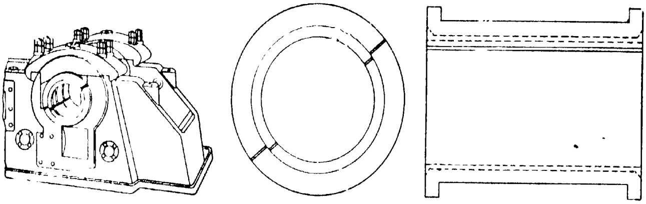


FIG. 142.—De La Vergne main bearing.

Two-piece Main Bearing for Vertical Diesels.—In vertical engines the wedgeless bearing, built along the lines of Fig. 143, is a common practice. The bearing, here shown, is provided with a ring oiling system; two rings are used, and the lubricating oil is carried in a cellar in the bearing housing. Much depends on the rings working freely. Ruined bearings frequently occur where the rings cease rotating, leaving the bearing without oil. With this bearing it is absolutely imperative that the oil in the cellar be maintained at a high level. To insure copious lubrica-

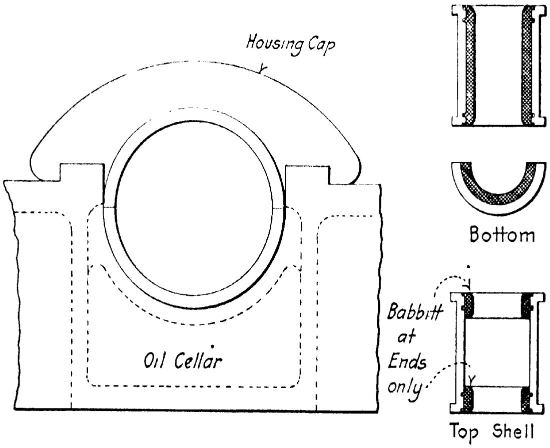


FIG. 143.—Main bearing McIntosh & Seymour Type A Diesel.

tion the distance between oil level and point of contact of ring and shaft should be as small as possible. It is self-evident that the greater this distance is made the smaller will be the amount of oil which will reach the shaft since there is a longer time interval to allow the oil to flow back down the ring. The same reasoning applies to the grade of oil used. If it is fairly free-flowing, the shaft will receive a meager supply. At the same time if the ring lies deep in oil the resistance may be enough to prevent the ring from rotating. The same type of bearing is adaptable to the use of a chain instead of a ring for oiling purpose. The chain should always be of brass in order to eliminate any grooving of the shaft. It is to be remembered that this brass chain will wear and break. If the bearings are not inspected occasionally, a broken chain may cause the babbitt to be ruined. It is an easy matter, when the engine is stopped, to pull a loop of the chain up through the bearing cap and run the entire length through one's fingers. Any links that are worn thin are easily detected, and, if many links are in a bad condition, the entire chain should be renewed. The advantage of the chain oiler lies in the greater amount of oil the links will carry to the shaft over that supplied by a ring oiler.

McIntosh & Seymour Type A Diesel Bearings.—The A-frame Diesel of this make is shown in Fig. 143. The bottom and top shells are not interchangeable since the top shell is only partly babbitted. Lubrication is by an oil cellar in addition to which a small amount of oil is supplied the bearing by a mechanical lubricator.

McIntosh & Seymour Type B Diesel Bearings.—The bearing of the Type B engine is of the same design as the Type A engine although the lubrication is by stream feed wherein a stream of oil is fed continuously to each bearing and no cellar is used, the oil dropping into the crankcase and from here flowing to an oil sump.

McIntosh & Seymour Marine Diesel Bearings.—Both the trunk and crosshead Diesels use a two-piece main bearing of the same general design as above but pressure oiling is followed.

Busch-Sulzer Diesel Main Bearing.—The same general type of bearing, Fig. 144, with a pressure oiling system is found on the Busch-Sulzer Type B and C Diesels. With this design the top and bottom shells are identical and, in emergencies, when the lower shell wears, the bearing may be reversed, using the top for the bottom. There is no wear on the top shell, and a worn bearing can be used here for a time. This practice should not be maintained continually since a loose fit at the top shell may cause the shaft to whip and lift upward, but by removing shims this clearance can be reduced.

With the pressure oiling system used on this bearing it is essential that there be a good seal between the shell and the housing. If not, the oil will escape at the ends, and the shaft will fail to receive proper lubrication. Ordinarily, if the oil passage to one bearing becomes clogged, the gage on the oil-pressure line will show an increased pressure. If another bearing is worn, thus allowing a freer passage than usual, it is possible for one line to be completely clogged without any pressure change being evidenced.

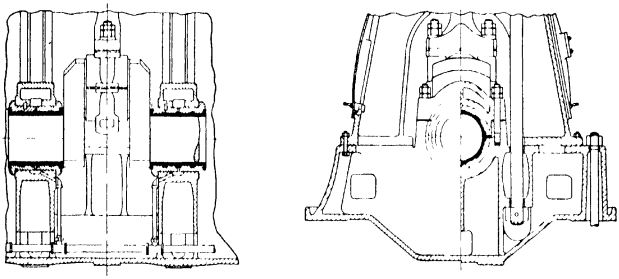


Fig. 144.—Busch-Sulzer Type B main bearing.

Fulton Diesel Main Bearing.—The Fulton engine uses a two-piece bearing of similar design. This is shown in Fig. 86. Lubrication is by means of stream lubrication through adjustable needle valves. The oil dropping into the crankcase is removed by a pump driven by the crankshaft and after cooling, is returned to the reservoirs above the bearings.

Dow Diesel Main Bearing.—The Dow main bearings are of the typical two-piece design. Lubrication is by rings dipping into oil cellars under each bearing.

Nordberg Diesel Main Bearing.—The three designs of two-cycle Nordberg Diesels use a two-piece main bearing; as shown in Fig. 72 *A*. Lubrication is by stream feed; the oil after leaving the bearings drops into the crankcase where it is picked up by a pump, sent through a cooler and filter into an overhead gravity feed-storage tank.

tion the distance between oil level and point of contact of ring and shaft should be as small as possible. It is self-evident that the greater this distance is made the smaller will be the amount of oil which will reach the shaft since there is a longer time interval to allow the oil to flow back down the ring. The same reasoning applies to the grade of oil used. If it is fairly free-flowing, the shaft will receive a meager supply. At the same time if the ring lies deep in oil the resistance may be enough to prevent the ring from rotating. The same type of bearing is adaptable to the use of a chain instead of a ring for oiling purpose. The chain should always be of brass in order to eliminate any grooving of the shaft. It is to be remembered that this brass chain will wear and break. If the bearings are not inspected occasionally, a broken chain may cause the babbitt to be ruined. It is an easy matter, when the engine is stopped, to pull a loop of the chain up through the bearing cap and run the entire length through one's fingers. Any links that are worn thin are easily detected, and, if many links are in a bad condition, the entire chain should be renewed. The advantage of the chain oiler lies in the greater amount of oil the links will carry to the shaft over that supplied by a ring oiler.

McIntosh & Seymour Type A Diesel Bearings.—The A-frame Diesel of this make is shown in Fig. 143. The bottom and top shells are not interchangeable since the top shell is only partly babbitted. Lubrication is by an oil cellar in addition to which a small amount of oil is supplied the bearing by a mechanical lubricator.

McIntosh & Seymour Type B Diesel Bearings.—The bearing of the Type B engine is of the same design as the Type A engine although the lubrication is by stream feed wherein a stream of oil is fed continuously to each bearing and no cellar is used, the oil dropping into the crankcase and from here flowing to an oil sump.

McIntosh & Seymour Marine Diesel Bearings.—Both the trunk and crosshead Diesels use a two-piece main bearing of the same general design as above but pressure oiling is followed.

Busch-Sulzer Diesel Main Bearing.—The same general type of bearing, Fig. 144, with a pressure oiling system is found on the Busch-Sulzer Type B and C Diesels. With this design the top and bottom shells are identical and, in emergencies, when the lower shell wears, the bearing may be reversed, using the top engine with a wedgeless bearing, the only way to raise the shell is by the insertion of shims between the housing and the shell or by scraping down the high bearings. Practically all manufacturers bore the housings true to a test bar. Figure 145 is self-explanatory as to the method of truing the housing, and it is only necessary to align the bottom of the housings; this done the bearing shells will be in correct alignment. It should never be necessary to shim up the bearings of a new engine. The work is usually performed in a painstaking manner, and seldom does any housing show more than a few ten-thousandths out of alignment.

FIG. 145.—Fitting main bearing shells to seats in baseplate.

The housings are filed and scraped until they test true. The bearing shells are all made of equal thickness, and, after being bedded in the housings, are checked for alignment by a test shaft. If any error appears, it is corrected by scraping of the babbitt.

After the engine is installed in a power plant, the bearings begin to wear. No matter how excellent the lubricating system may be, the wear starts as soon as the engine turns over. Since the babbitt linings in the several bearings are never of equal density the wear is by no means uniform. Varying pressures in the engine cylinders tend to increase the discrepancy in the rate of bearing attrition. If the flywheel weight is supported or partly supported by one of the end main bearings, this bearing will probably show a more rapid wear than the others. Oil-engine operators should recognize these facts and take proper steps to maintain the bearing alignment. Too many allow matters to drift until one or more bearings start to run hot. In most instances it is then too late to save the babbitt lining.

Inspection of Bearings.—Just as there should be a regular schedule in grinding valves and pulling pistons, the engineer should test the bearings at stated intervals. It is difficult to give a rule as to the length of time elapsing between inspections; the operator is not unduly active if he arranges to examine the bearings every 6 months. To remove the bottom half of the bearing, it is only necessary to lift off the top half and turn the flywheel. The revolving shaft will draw the lower bearing from the housing. The thickness of the shell should be measured at both ends and at the center; several measurements should be taken at various points. If these measurements are the same, it is evident that the wear on the particular bearing has been uniform. If the measurements check with readings taken on the other bearings, the main bearings are in alignment. If one bearing shows that it is low, the engineer must decide his course of action. It is possible to shim up under this low bearing until it is as high as the other. If the difference is marked this is the best procedure; sheet steel makes the best liners. If the variation is less than 0.005 in., the best scheme is to scrape the other bearings, removing an amount of babbitt to compensate for the variation between the highest and lowest bearings. In cases where all the bearings vary, scraping is the only remedy.

FIG. 146. Welding of fractured crankshaft.

That more attention should be given to bearings is evidenced by a crankshaft failure, photograph of which is here reproduced in Fig. 146. On examination the shaft showed marked crystallization. When the alignment of the bearings was checked one was found to be $\frac{1}{32}$-in. lower than the others. The shaft had been unsupported between two of the cranks, and at each power stroke of these two cylinders it had bent. This bending, continuing for months at the rate of 350 times per minute, resulted in crystallization and final failure. In this breakage a loss of some $7,000, plus loss of operating time, is traceable to misalignment of the main bearings.

Measuring Bearing Wear.—A ready means of measuring the amount the shaft is below its original position is available if the builder has supplied a bridge gage. Some builders at the time the engine leaves the shop stamp the bridge-gage readings on the engine bed near each bearing and supply the engine buyer with the gage with which these measurements were made. This gage is similar to the one shown in Fig. 147, and the measurements recorded are the distances between the top of the shaft at each bearing and the point on the gage. Any subsequent measurements with the same gage, when compared with the original measurements, show how much the shaft is down at each bearing.

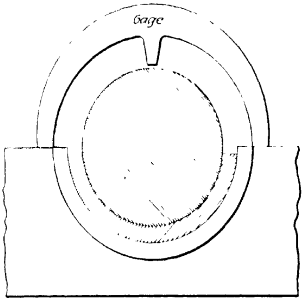


FIG. 117.—Bridge gage to check clearances.

Variation in Bearing Wear.—The variation in the bearing levels should not exceed 0.001 in., for the housings are bored with a tolerance equal or less than this amount. If more than this, the shaft will likely spring in operation. On engines with the flywheel mounted on an extension shaft the wear is more rapid on the main bearing adjacent to the flywheel. The thoroughly versed erector always sets this bearing from 0.003 to 0.004 in. higher than the other bearings. This allows a wear of 0.006 to 0.008 in. before the bearing becomes too low. Because of the initial misalignment this particular main bearing always runs warmer than the other bearings. Many operators experience a great deal of worry over the temperature condition of this bearing when, in fact, the higher temperature actually indicates that the bearing is in proper shape. When the temperature becomes lower than with the other bearings, it can be taken as evidence that this bearing has worn low and must be raised.

Measuring Bearing Clearance.—For measuring the clearance in main bearings, where no gage is provided, the top cap of the bearing being worked on is lifted off with chain falls. The leads or soft fuse wires, say $\frac{1}{32}$ in. thick, are laid circumferentially or around the top of the shaft, after which the cap is replaced and the nuts sledged up to their former position. On engines with shafts 9 in. or over, three leads, one at each end and one in the center should be used. The cap is removed a second time and the thickness of the leads measured. The leads taken indicate the total amount of clearance between the shaft and the babbitt lining of the top half of the bearing, but they give no information as to how much of this clearance is due to wear of the bottom half and how much to wear of the top half. For this reason it is necessary that the lower bearings be measured for wear as indicated above. The clearance should be about 0.001 in. per inch of shaft diameter.

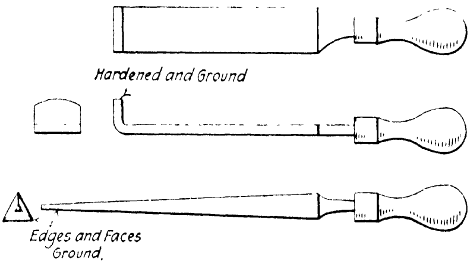


FIG. 148.—Bearing scrapers.

Refitting a Bearing.—There are many instances where a bearing becomes hot, causing the babbitt to drag and cut; yet in the majority of cases it is not necessary to rebabbitt this bearing, for the actual reduction in bearing thickness may be less than 0.002 in. The proper procedure is to use scrapers made of old files along the lines of Fig. 148, although there are various scrapers on the market, which, if available, are probably handier.

The babbitt should be smoothed down and all loose and damaged metal cut out. This may cause part of the liner to be so low that it fails to contact with the shaft. This is not objectionable since the ordinary shell has an excess of bearing surface. Where the damage is considrable, a new babbitt liner is the only solution.

With very large engines short mandrels are usually supplied, one each of the same size as the main bearings, crankpins and crosshead pins. When refitting a box that has been badly worn, overheated, or rebabbitted, the work can be done at any convenient place in the engine room, using the mandrel for spotting in. This makes it unnecessary to do the spotting on the bearings itself and does away with a lot of laborious handling of the box within the restricted space in the housing.

Rebabbitting a Bearing.—When the situation arises, the engineer, must determine the best way of rebabbitting and getting the unit back into service. In too many cases the old babbitt is melted out and a new lining run with a mandrel of the same diameter as the bearing. Often but not always the babbitt is scraped to a fair fit to the journal.

It has been generally found that, when handled in the manner outlined, the babbitt fails to maintain an anchor to the bearing shell; it has a decided tendency to loosen and finally crack when in service. Usually, the particular make of babbitt is blamed, when almost invariably the methods followed in the babbitting process are at fault.

In preparing to run a new babbitt liner, the old metal should be carefully removed. As a rule it is necessary to heat the shell in order to remove the babbitt from the dovetail grooves. After the metal is removed, the shell should be thoroughly cleansed of any oil deposits by first washing with gasoline and then with lye water and wiping dry.

Muriatic acid is next placed in a porcelain or pottery vessel, such as a small crock. Zinc scraps should be dropped into the acid until it is thoroughly "cut" or neutralized. The bearing shell is then brushed rather heavily with the cut acid. A blowtorch is played against the side of the shell while the engineer rubs a solder stick all along the inner surface of the shell, as shown in Fig. 149. To insure a thorough tinning of the shell, sal ammoniac should be sprinkled over the surface as the coating of tin solder is rubbed into all the grooves. The tinning process insures a positive anchorage of the babbitt when run to the bearing shell.

Without the preliminary tinning the babbitt, on cooling, appears to shrink away from the shell.

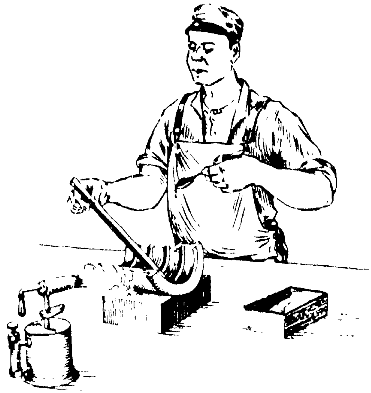


FIG. 149.—Tinning bearing shell before babbitting.

Before pouring the babbitt, it is necessary to procure a mandrel, if the engine builder has not supplied one. If a more improved device is not at hand, a quite serviceable one can be made of a piece of pipe and a companion flange, as outlined in Fig. 150 at *A*.

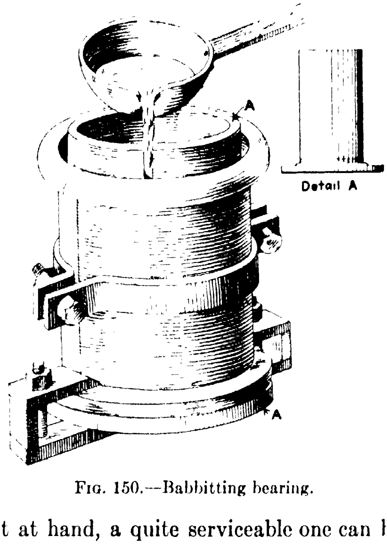


FIG. 150.—Babbitting bearing.

The pipe should be at least one-eighth inch smaller in diameter than the shaft journal. A piece of asbestos should be placed on the flange and the two bearing shells clamped together with paper or brass shims between the halves and slipped over the mandrel. To insure against leakage, wet fireclay may be placed around the bottom edge of the shells.

Only a high grade of babbitt metal should be used on engine bearings. In heating the babbitt in a ladle, two blow-torches impinging against the ladle are better than a coke or coal fire. The metal should not be hot enough to burn a pine splinter when the latter is thurst into the molten babbitt. If the stick chars but does not blaze up, the metal is about right for pouring.

After the bearing has been poured, it is never advisable to peen the surface of the babbitt. Although this practice is quite usual, the peening does not make the babbitt anchor any better and often the hammering loosens the bond between the babbitt and shell.

Boring Bearing.—Any bearing that is under high pressures such as occur with an engine bearing, should be bored to exact diameter. This is by no means difficult if a lathe or boring mill is at hand. Figure 151 illustrates one method of boring the bearing. Here the shells are checked on the lathe and the boring tool held in the toolpost. The lathe speed must be low or chattering will occur. Almost all bearings have a flat surface at the ends, and

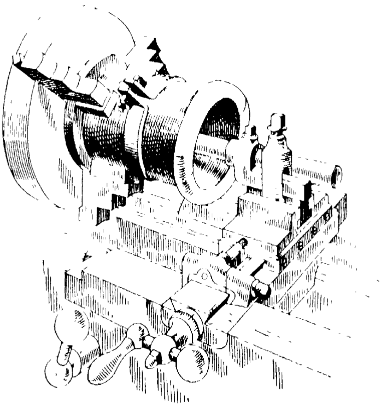


FIG. 151.—Boring rebabbitted shell.

the bearing must be bored square with this edge. Care must be used in chucking the shells to get it in the correct position.

If the bearing is too heavy to be chucked, it can be clamped to the lathe carriage and a boring bar made of a piece of steel shafting with a cutting tool passing through the shaft and held fast by a setscrew. Often a drill press is called into service. In this case the shells are clamped on the press table and a boring tool made with two cutting edges to prevent side slippage of the spindle.

After boring, the shells should be separated and at the edges the babbitt should be cut away in order to provide a clearance which will enable the oil to enter between the bearing and shaft. All bearings on engines should be scraped to an exact fit to the journal. This is best done by coating the journal surface with Prussian blue or lampblack, in a pinch red lead thinned down with gasoline will do. If possible, a mandrel exactly the diameter of the shaft should be secured. The spotting of the bearing can be done with the mandrel, thus avoiding the difficult handling of the bearing in and out of the engine crankcase. A very light coating is applied and the shell placed on the journal and lightly rotated. When removed, the shell will be coated with the lampblack at those points where the babbitt is high. Scraping after repeated trials will finally cause the babbitt to bear evenly at all points. In cutting oilways in the babbitt, the grooves should not extend to the ends of the shell nor to the chamfer at the parting edges, this is to prevent a too rapid flow of oil out of the bearing. In fact, all that is needed are two short grooves, intersecting at the middle of the bearing and about four inches in length.

With engines where the lower shell rests on a bored seat in the housing, the shell, including the babbitt, must be of the same thickness as the first shell when new. Consequently, in boring this must be measured and in scraping the babbitt must be cut away to give this exact value. If the procedure outlined is followed, a bearing equal to the original one will result. A small amount of care and patience in doing the work will bring large returns in long bearing service.

Hot Bearings.—A hot bearing is a trouble that every engineer some day will experience. Generally this occurs when the load is heavy and when a shut-down is an impossibility. The engineer must maintain his presence of mined even though the bearing smokes. It is an all too common practice for the operator excitedly to douche the bearing with a bucket of water. This is of no avail. Water is at best a poor lubricant, and it will merely make matters worse by washing off the little oil that does cling to the shaft. The water strikes the shell and the shaft; this results in the contraction of the shell, thereby loosening the babbitt. Another bad practice is the use of an air hose in a vain endeavor to cool the bearing. The only correct procedure is to run an oil pipe or hose to the bearing and feed a heavy stream of cool oil through the inspection hole in the cap directly on to the shaft. If the bearing is provided with an oil cellar, the drain cock to this should be opened and the oil allowed to flow out after passing over the shaft. It is best to run the engine light until the bearing cools off. If the engine is arranged to allow a cylinder to be cut out, the two cylinders adjacent to the hot bearing should be operated idle, with the exhaust valves blocked open. This relieves the damaged bearing of part of the pressure due to the cylinder explosion. After shutting down, the bearing liners should be examined and any necessary repairs made.

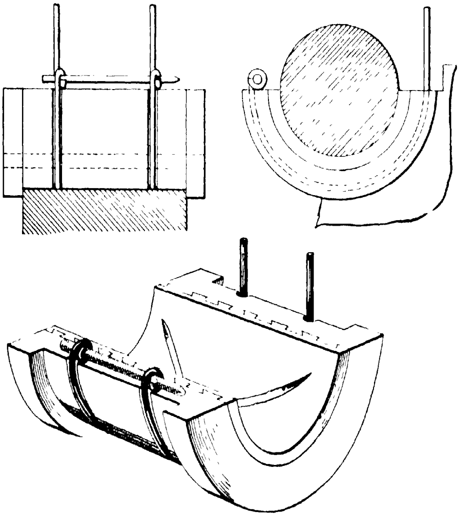


FIG. 151A.—Removing bearing.

Withdrawing Bearing Shells.—It is rather difficult to remove the lower half of a main-bearing shell without disconnecting the

the bearing must be bored square with this edge. Care must be used in chucking the shells to get it in the correct position.

If the bearing is too heavy to be chucked, it can be clamped to the lathe carriage and a boring bar made of a piece of steel shafting with a cutting tool passing through the shaft and held fast by a setscrew. Often a drill press is called into service. In this case the shells are clamped on the press table and a boring tool made with two cutting edges to prevent side slippage of the spindle.

After boring, the shells should be separated and at the edges the babbitt should be cut away in order to provide a clearance which will enable the oil to enter between the bearing and shaft. All bearings on engines should be scraped to an exact fit to the journal. This is best done by coating the journal surface with Prussian blue or lampblack, in a pinch red lead thinned down with gasoline will do. If possible, a mandrel exactly the diameter of the shaft should be secured. The spotting of the bearing can be done with the mandrel, thus avoiding the difficult handling of the bearing in and out of the engine crankcase. A very light coating is applied and the shell placed on the journal and lightly rotated. When removed, the shell will be coated with the lampblack at those points where the babbitt is high. Scraping after repeated trials will finally cause the babbitt to bear evenly at all points. In cutting oilways in the babbitt, the grooves should not extend to the ends of the shell nor to the chamfer at the parting edges, this is to prevent a too rapid flow of oil out of the bearing. In fact, all that is needed are two short grooves, intersecting at the middle of the bearing and about four inches in length.

With engines where the lower shell rests on a bored seat in the housing, the shell, including the babbitt, must be of the same thickness as the first shell when new. Consequently, in boring this must be measured and in scraping the babbitt must be cut away to give this exact value. If the procedure outlined is followed, a bearing equal to the original one will result. A small amount of care and patience in doing the work will bring large returns in long bearing service.

Hot Bearings.—A hot bearing is a trouble that every engineer some day will experience. Generally this occurs when the load is heavy and when a shut-down is an impossibility. The engineer must maintain his presence of mined even though the bearing smokes. It is an all too common practice for the operator back whereupon the magnetic pull again starts the lengthwise motion.

A second cause is the existence of a low bearing which will set up a lengthwise travel of the shaft. A third cause lies in the existence of a crankpin box which is incorrectly fitted to the pin, throwing the plan of travel of the connecting rod out of square with the shaft. The rod thus exerts a lengthwise thrust upon the pin and shaft. The engineer should see if the cause can be eliminated before using the thrust ring mentioned.

Bearing Pressure.—The dimension of the main or shaft bearings are almost entirely dictated by the shaft size, which in turn depends upon the cylinder dimension and number of cylinders, etc. As a general statement, it may be said that the bearing pressures should not exceed 700 lb. per square inch of projected area.

Comparison of Two-cycle and Four-cycle Bearing Wear.—In a four-cycle engine the downward pressure upon the lower bearing shell is exerted during only part of the cycle. Upon the suction stroke and for the later part of the exhaust stroke the unopposed inertia of the piston is sufficient to eliminate or at all events greatly reduce the downward thrust on the bearing.

The film of lubrication has, then, on the four-cycle engine a chance to reform and consequently bearing wear due to poor lubrication is not prevalent.

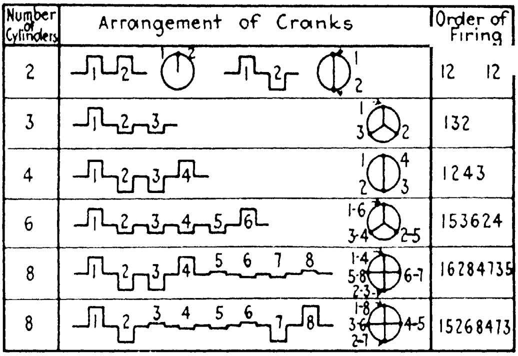
In the two-cycle, single-acting engine the downward thrust exists during the entire cycle. If the lubrication is not positive there is danger of the film of oil breaking. In this event metal to metal contact would occur, resulting in rapid bearing wear. Where pressure or stream lubrication is carried out, no more difficulties are experienced on the two-cycle than on the four-cycle engine. The problem of bearing wear in the two-cycle engine was solved when a good system of lubrication was adopted.

CRANKSHAFTS

The crankshaft is rightly one of the most important parts of the engine. The life of the Diesel depends in a great measure upon the proper design and maintenance of the shaft.

In the small and medium powered engine, say, up to engines of 22-in. diameter, the shaft is a one-piece forging of open-hearth steel of around 64,000 lb. tensile strength. In larger cylinder units the shaft is necessarily built-up since it is well nigh impossible to secure such large forging.

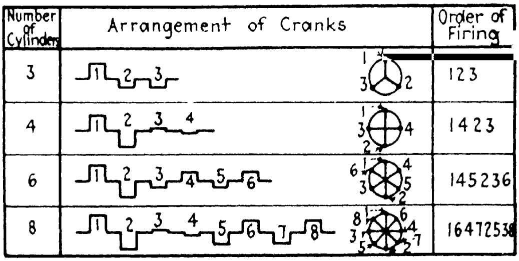
The arrangement of the cranks and order of firing of the cylinders are shown in Fig. 153 for four-stroke-cycle engines and in Fig. 153 *A* for two-stroke cycle units.



Number of Cylinders	Arrangement of Cranks	Order of Firing
2		12 12
3		132
4		1243
6		153624
8		1628473
8		15268473

FIG. 153.—Order of firing, four-stroke-cycle engines.

The theory entering into the design of a Diesel crankshaft is no different from that of steam engine, gas engine, or air compressor shafts and is given in all treatises on pure design. Those



Number of Cylinders	Arrangement of Cranks	Order of Firing
3		123
4		1423
6		145236
8		1647253

FIG. 153*A*.—Order of firing, two-stroke-cycle engines.

who are interested are referred to such volumes, for the complete treatment is beyond the scope of this work and is a matter of design rather than purchase and operation.

Lloyd's Rule for Crankshafts.—Lloyd's survey demands that the crankshaft diameter be as determined by the formula

$$D = \sqrt[3]{D_c^2(AS + BL)}$$

When D = shaft diameter in inches.
D_c = cylinder diameter in inches.
L = distance below bearing centers, inches.
S = piston stroke, inches.
A = constant
B = constant } as per table below.

Number, cylinders	Compressor driven by overhung crank		Without compressor	
	A	B	A	B
1, 2, 3	0.086	0.038	0.089	0.037
4	0.093	0.037	0.099	0.036
5, 6	0.103	0.035	0.111	0.035
8	0.120	0.034	0.131	0.033

The table is based on a four-stroke-cycle single-acting engine If the engine is two-cycle, the constants for a given number of cylinders are those constants shown in the table for twice the given number of cylinders.

United States Steamboat Inspection.—The requirements of the United States Steamboat Inspection Bureau are given in Chapter XXIV covering the general rules of the Inspection Service. In the same chapter are given the rules of the American Bureau of Shipping.

Crankshaft Proportions.—Some designers proportion crank webs by the formula

$$a = \sqrt{\frac{3LD}{4bF}}$$

where a = thickness of web in the direction of the pin and shaft length.
L = load on crankpin.
D = ½ distance between web centers.
b = web breadth.
F = allowable fibre stress = 7,500 lb. per square inch.

In a discussion before the English Diesel Engine Users Association it was recommended that, using the cylinder bore as unity, the crank dimension should be approximately

Diameter of pin and shaft	0.525 to 0.54
Length of bearings	0.75 to 0.80
Length of crankpin	0.525 to 0.54
Thickness of web	0.32
Width of web	0.8

Fractured Crankshafts.—The bogie of the Diesels when first introduced in this country was broken crankshafts. Undoubtedly the one thing that proved an obstacle to the introduction of the oil engine was this question. There were a few isolated cases of fractured shafts. These without reservation were due to one of two things. The early Diesel operator knew but little about the engine and believed in letting well enough alone. No attempt was made toward the adjusting of the various parts. The fuel valve frequently got out of order and opened too early, causing preignition. These excessive pressures had to be relieved by some means. Often the head gave way, but at times the head proved stronger than the shaft, and so a fractured shaft was the result. The second cause was the failure to take up the wear in the main bearings. Frequently the inside bearings became worn, allowing the shaft to be supported by the two outside bearings only. This produced a deflection in the shaft which was repeated and reversed each revolution. Ultimately the shaft gives way. As a rule the break occurs between the pin and web or throw. More liberal fillets at this point along with more knowledge acquired by the engineer will eliminate this danger.

Welding of Crankshafts.—It has been quite customary in the past to discard a shaft upon fracturing. When it is considered that a 9-in. shaft for a four cylinder 500-hp. engine costs around $7,000, it is apparent that it should not be junked if it can be saved.

Of late years electric, acetylene and Thermit welding have been used. Of the three, apparently electric and Thermit are uniformly successful. In Fig. 146 is shown a 500-hp. engine crank which broke due to a low bearing. Thermite welding was resorted to with success. Such work should be left to experts and if done right is economical. In the case in question the cost was $1,000, a saving of $6,000 over a new shaft.

Scored Shafts.—There are occasions when an engine crankshaft becomes scored to a serious extent or worn unevenly at the journals. The average engineer is prone to think that a shaft in either of these conditions is worthless and should be removed at once. When the damage is severe, of course this is necessary. More often a little work will allow the engine to still pull its load. In cases of scored shafts an emery stone will serve to smooth up the shafts, finishing with a scraper and a final polish by lapping. Figure 154 outlines a lapping sleeve used by one large Diesel user. The scores do not cause trouble providing the edges of the cuts are smoothed off, preventing the babbitt from being picked up. When the wear is uneven, the bearing shell should be scraped to a fit, even though the diameters at the two ends vary to a marked degree.

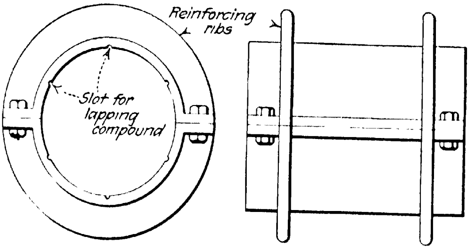


FIG. 154.—Lapping sleeve.

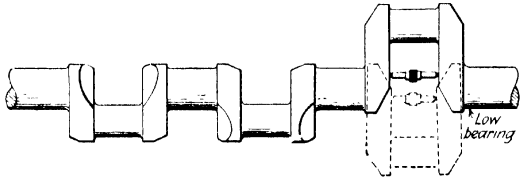


FIG. 154A.—Checking shaft flexure.

Checking Flexure of the Shaft.—It is highly important that at least once every six months the shaft should be examined for flexure. This is done by using an inside micrometer and by measuring the distance between the crank webs both when the crank is at top and at bottom centers, as shown in Fig. 154 *A*. If there is any flexure, it will show a change in the measurement. When these measurements are being made, the pistons and rods should be kept coupled up. If not, the flywheel might cause the shaft to bow slightly, and we are interested in the shaft condition when all parts are connected. In overhauling or erecting an engine, the measurements may be made before the flywheel or rods are connected.

Bearing Wear.—The flexure of the shaft will gage the amount of bearing wear or rather the amount of inequality in the bearing wear. In case of a four-cylinder four-cycle engine reported to the Diesel Engine Users Association, the following measurements were recorded after 18,000 hours of operation.

Cranks	1 Inches	2 Inches	3 Inches	4 Inches
Top center	7 118	7 099	7 1050	7 114
Bottom center	7 116	7 0985	7 1015	7 117
Differences	0 002	0 0005	0 0005	0 003

One point, however, should be remembered—the presence of shaft flexure, as indicated by a difference in the top and bottom measurements of a crank pin, does not locate the low bearing. This must be found by measuring the bearing thickness.

Crankshaft Wear.—The amount of wear on pins and journals depends upon the character of the steel, the type of bearing, method of lubrication, and general care. All cranks and journals tend to wear oval since the high pressure on the piston occurs during a small portion of the crank circle. One 6½-in. shaft after 26,000 hr. service showed the following amount of wear, using the journals as numbered in Fig. 153.

No. 2 journal	oval 0 0015 in.
No. 3 journal	oval 0 0033 in.
No. 4 journal	oval 0 0011 in.
No. 5 journal	oval 0 0007 in
No 6 journal	oval 0 0018 in

CHAPTER XII

ADMISSION AND EXHAUST VALVES

TYPES. ADJUSTMENTS. REPAIRS

Valves.—The four-stroke-cycle Diesel engines are provided with air admission and exhaust valves as are all four-stroke-cycle gas engines. In fact, a number of Diesels are built with cylinder heads and valve mechanisms that are but slight deviations from the similar parts of a gas engine. This applies particularly to the horizontal Diesel of which a number possess what might well be termed "gas-engine heads." In the vertical engines the entire valve equipment is distinctly a Diesel feature, differing in many respects from any of the gas-engine designs.

In all Diesels the admission valves seat in valve cages along the lines of Fig. 157; the builders are not so unanimous as to the exhaust valve. This latter valve is some small horizontal engines is provided with a cage; in other makes the valve seats directly on the cylinder casting; in others a removable seat which is fastened to the head casting is supplied. Strictly speaking, the exhaust rather than the admission valve should be fitted into a cage. The hot exhaust gases, in passing through the valve opening, wear the ground seats very rapidly, while the admission valve is not subject to such an erosive action. From the operator's viewpoint the exhaust valve should be caged, as well as the admission valve. Engine builders, in a desire to eliminate the difficulty of lowering the exhaust cage under the cylinder head, place the admission valve in a cage and design the exhaust valve with a diameter to allow the latter to be lifted up through the admission valve cavity after the admission valve assembly has been removed. This enables the engineer to grind the exhaust valve by inserting a pin wrench through the admission opening.

In the horizontal engines where the valves are placed with the stems in a horizontal position, as well as in the vertical engines, usually both valves are caged.

Valve Camshafts and Levers.—The present-day vertical Diesels of American manufacture have camshafts along the cylinder heads and are driven by a vertical shaft from off the crankshaft. The drive is mainly by helical gearing. The camshaft carries the cams for the admission, exhaust, injection and starting valves; the cams are usually of cast steel, with a tool-steel nose on the fuel-injection cam. The lever mechanisms are quite similar for all makes of vertical engines, the variation being mainly in the arrangement for operating the starting valve.

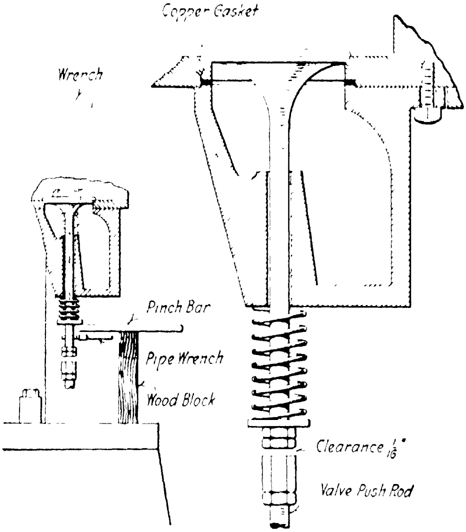


FIG. 155 -American Diesel exhaust valve.

The horizontal engines are all fitted with a layshaft along the engine frame which is driven on a 2 to 1 ratio from the main shaft. The valves receive their motion from cams or eccentrics mounted on the layshaft.

American Diesel or Busch-Sulzer Type A.—The former engine, which was the forerunner of the Busch Type A engine, is of interest since many are still in operation. The positions of the admission and exhaust valves are shown in Fig. 13. The camshaft is mounted within the enclosed crankcase and is driven from the main shaft, through an idle or intermediate gear, and operates at half engine speed. The exhaust valve seats vertically in the lower face of the head and is driven directly by a vertical push-rod pinned to the valve lever *B*. The lever is closed by a spring which is held in place by a washer and lock-nuts as shown in Fig. 155. This valve has two tapped holes in the top which allow a wrench to be used, but the size of the cap screws are such that it is often impossible to hold the valve with the wrench when disassembling on account of the burnt threads. The most satisfactory procedure to follow is outined in the sketch to the left of Fig. 155. The spring is compressed by a pinch-bar, allowing the engineer to grasp the valve stem with a pipe wrench. By placing an open-end wrench on the lock-nuts the nuts can be removed, whereupon the spring is released and the valve withdrawn. The valve has a 45-deg. angle to its face seating directly on the cast-iron base of the cylinder head. If the engineer desires to regrind, the valve is reinserted on the stem housing and a light spring set below the valve as indicated by the dotted outline.

The valve-stem housing is not bushed and in time wears enough to emit part of the exhaust gases. This can be eliminated by the reaming of the housing and the insertion of a brass tube of the correct diameter and bore. The exhaust pot has a water drip connection for the purpose of maintaining the temperature at a reasonably low value. A drip cock is also a useful adjunct to the pot since it will free the chamber of tarry deposits. The junction of the pot and cylinder head is sealed by a copper gasket. It is absolutely necessary that the stud bolts have clean threads for they are rather inaccessible, and if the stud nuts are not tight the gasket will blow.

Admission Valve.—The admission valve is contained in a cage, Fig. 156, and is actuated by a cam, similar to the exhaust cam, through the lever and push-rod shown. The cage has a gasketed joint at *a*. In placing the copper gasket on the joint and setting the valve cage into position, care .is to be exercised to prevent the gasket from wedging.

Adjustments.—The valve lever, since the clearance between its end and the valve nut is large, delivers a hammer blow to this nut and the fibre washer. The washer wears rapidly while the nut gradually shears the threads on the valve stem. The valve stem can be turned down and threaded to take a smaller diameter nut; a second remedy is the employment of a cotter pin through the worn nut and valve stem. The bushings on the lever pins also wear rapidly; these can be replaced by new bushings. The pins frequently become flat, and new pins of machine steel, case-hardened, can be turned to replace the defective ones. If the pins are oiled, as they should be, little wear will occur; while the pin bearings have only oil holes, small oil cups are far more serviceable.

In regrinding this valve the unit is disassembled. The valve is reinserted in the cage and the dashpot again placed on the valve stem. This constrains the valve stem to remain in its proper position while the valve face is ground to the correct angle.

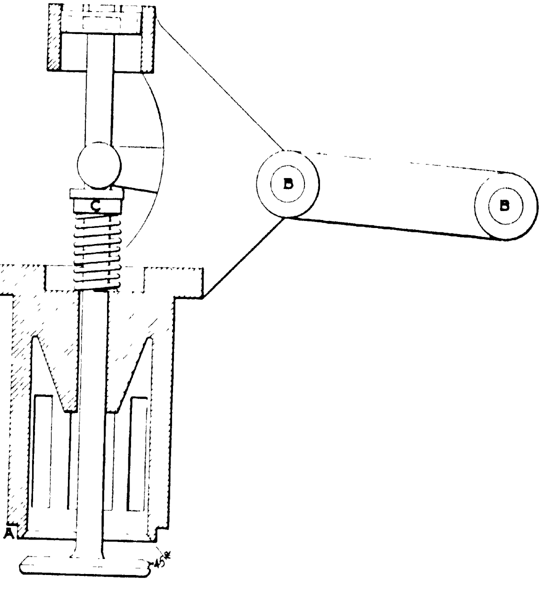


Fig. 156.—Admission valve cage, American Diesel.

Adjusting Valve Levers.—The long nut on the upper end of the exhaust valve push-rod is screwed on until it clears the valve stem nuts by 1/16 in. when the valve is at rest and the push-rod is in its lowest position. The lock-nuts on both push-rod and valve stem must be jammed tight to prevent any change in this clearance when the engine is running. The upper end of the admission valve push-rod is adjusted to allow the valve end of the rocker to clear the fibre washer by 1/32 in. when the valve is seated.

A layout of the valve mechanism is shown in Fig. 13. This also includes the fuel valve rocker arm and rod. The cams are of cast steel keyed to the camshaft. Change in the timing of the cam assembly can be effected by shifting the cam gear a tooth or two. This is made possible by sliding the idle gear until it is out of mesh with the cam gear. Alteration of an individual cam is obtained by the employment of an offset key which will move the cam the required amount if the adjustable cam nose does not give enough adjustment.

Valve Timing.—Figure 156*A* gives the timing of admission, exhaust and fuel valves on the 225-hp. American-Diesel engine.

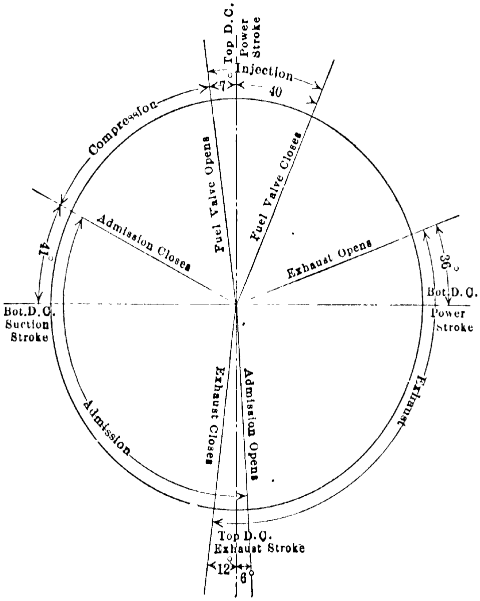


FIG. 156*A*.—Valve timing of American Diesel.

These timings represent standard practices and are equally applicable to 120- and 170-hp. engines of this make. In this diagram each quarter circle represents 180 deg. and not 90 deg., being equal to one complete stroke of the engine piston. The degrees indicated are to be laid off on the fly wheel rim and are not percentages of the piston travel. To translate these degrees to inches on the periphery of the flywheel is a matter of simple mathematics. For example, if the wheel be 8 ft. in diameter, the circumference is then 302 in. One degree is in this case $^{320}/_{360}$ – in. or 0.83 + in. If the operator uses a value of 0.8 in. per degree, he will be working amply close.

Busch-Sulzer Type B Diesel Valves.—In exterior form the admission and exhaust valves and cages are quite similar. Both are placed vertical in the cylinder. The interior arrangement of the cages, however, are by no means identical. The exhaust-valve cage has a removable valve-stem bushing; the cavity between this bushing and the cage body constitutes the cooling-water jacket. The admission valve cage has the valve stem support cast integral with the cage proper. Both cages are provided with removable seats as appears in Fig. 157. This feature is of importance to the operator since it obviates the necessity of replacement of the complete cage when the valve seat has been worn excessively.

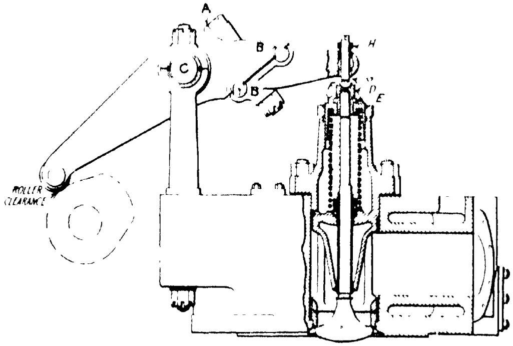


FIG. 157.—Busch-Sulzer Type B exhaust valve and rocker.

The valves, both admission and exhaust, have cast-iron bodies with steel stems. In disassembling the valves, the lock-nuts *D* and *E* are unscrewed, the valve being held by a pin wrench inserted in the two pin holes in the valve body. The removal of the spring cap *F* allows the valve to be withdrawn. The exhaust-valve seat, of course, experiences the greatest wear; consequently frequent refacing of the cage seat becomes necessary. After a number of refacings with a reamer the cage seat lies too deep in the cage; when this occurs, a new removable seat ring can be obtained.

Cam Levers.—The valves are actuated by levers which receive their motion from a camshaft that lies along the cylinder head. This camshaft is driven by the engine shaft through the intermediation of the vertical governor shaft and two sets of helical gears.

To allow the valves and cages to be lifted without the removal of the entire rocker arm and shaft, the rockers are in two parts. These two parts are held together by one bolt *A* while two dowel pins *B* preclude the possibility of the parts being improperly aligned on reassembly. One end of the rocker carries a hardened steel roller while the other end is fitted with the adjusting screw *G*, which screws into a steel pin and is locked by the nut *H*.

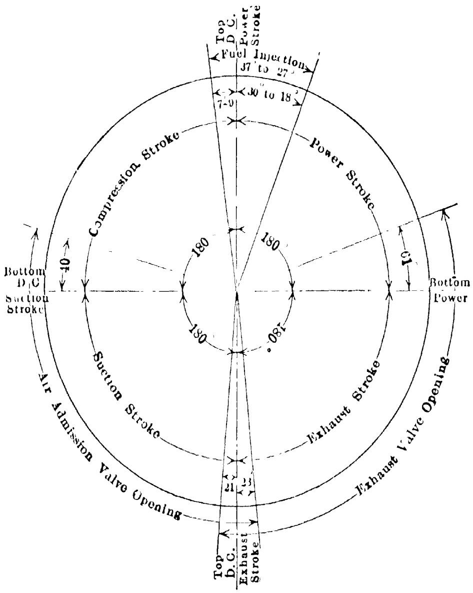


FIG. 158.—Busch-Sulzer 500 hp. Type B four-cylinder, 200 r.p.m. Diesel valve timing diagram.

Valve Lever Clearances.—The adjusting screw enables the operator to obtain the required clearance between the roller and the cam when the valve is closed. The rocker arm is held against the valve stem, and the clearance is measured between roller and cam by ordinary thickness gages. The clearance valves that are correct for all sizes of the Busch-Sulzer Type B engines are as follows:

Admission valve cam clearance	0 012 in.
Exhaust valve cam clearance	0 018 in.
Fuel valve cam clearance	0 004 in.
Starting valve cam clearance	0 012 in.

Since the clearances are established when the engine is cold, it becomes necessary to note whether the valves close after the engine has become warmed. If not, the adjusting or push screws can be backed off to give these values. Figure 158 is the usual valve timing on this engine.

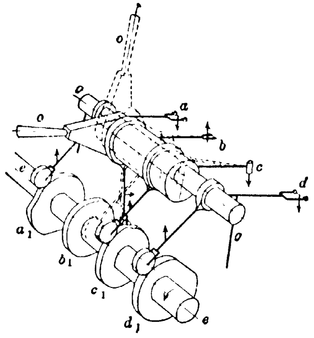


Fig. 159.—Busch-Sulzer rocker layout.

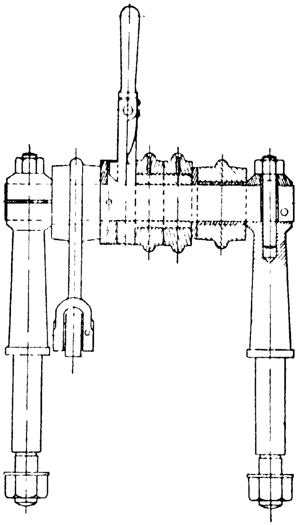


Fig. 160. -Lever shaft

Camshaft Layout.—Figure 159 is a schematic outline of the actuating valve levers of the Busch-Sulzer engine. The lever shaft *g* is supported by the pedestal bearings, better shown in Fig. 160. The lever *d* controls the exhaust valve, while *a* actuates the air admission valve, and the lever *b* governs the fuel valve, the lever *c* opening the air starting valve in starting the engine. The levers *b* and *c* are mounted on an eccentric bushing which is supported by the shaft *g*. In starting, the lever *o* moves the eccentric bushing to the "starting" position where the fuel lever *b* does not engage its cam *b*; this movement brings the starting lever *c* into contact with its cam *c*. This action is carried out on the two starting cylinders; on the two idle cylinders the eccentric handle is moved to the neutral position, cutting out the fuel valve. It should be understood that only the two starting cylinders have the starting valves.

McIntosh & Seymour Type A Diesel Valves.—The admission and exhaust valves of this engine follow the form shown in Fig. 161. Both valves have the same dimensions, the difference between the two being the removable seat used on the exhaust valve; the valve seats have a 60-deg. angle with the stem. This seat is an alloy-steel ring held in place by two countersunk screws.

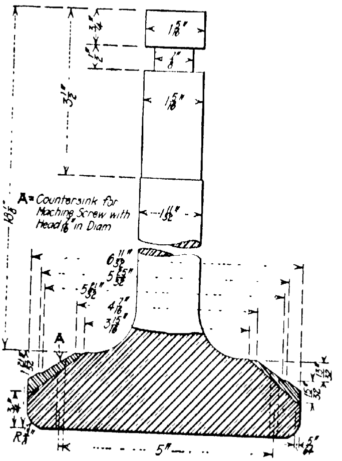


FIG. 161.—McIntosh & Seymour exhaust valve.

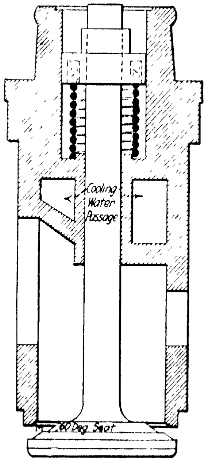


FIG. 162.—Exhaust valve and cage.

In renewing this ring it is very essential that the surfaces be thoroughly cleaned. Occasionally the ring is slightly distorted when received; it must be ground true before it is placed on the valve. The exhaust and admission valve cages are of the same external dimensions. They differ only in the exhaust cage having a water cavity around the valve stem. The cages rest in machined cavities in the head, the joint at the point *a* being sealed with a copper gasket. The gasket must be of the proper width to eliminate danger of cramping when the cap is placed over it. Figure 162 shows the exhaust valve and its cage.

To disassemble the valve, the pin *B*, Fig. 163, is driven out of the valve stem, and the bushing *a* is removed. The spring cap *D* is pushed downward, allowing the slotted ring *C* to be slipped from around the valve stem. This being done, the spring cap and the spring are withdrawn from the top of the cage while the valve can be removed through the bottom end of the cage.

Camshaft.—The camshaft lies along the cylinder heads, resting in bearing pedestals mounted on extensions from the cylinder castings. The camshaft is driven, through helical gears and a vertical governor shaft, by the main engine shaft.

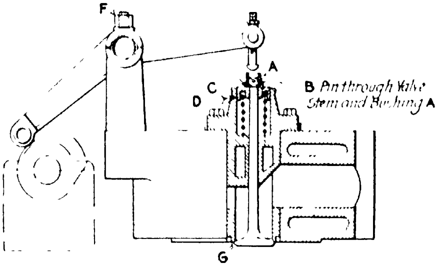


FIG. 163. McIntosh & Seymour valve layout.

Figure 164 shows the valve rocker arrangement where *d* is the exhaust rocker, *a* is the admission rocker, *b* the fuel rocker, and *c* the air-starting rocker. The cams for *b* and *c* are in one piece; the engine is started by shifting the starting roller *c* until it contacts with the starting cam; a small lever is provided on the rocker

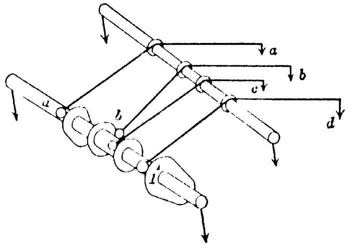


FIG. 164.—McIntosh & Seymour valve-rocker layout.

for this purpose. After the engine fires, the roller *c* is moved out of contact with its cam.

In removing the valve cages, the caps of the pedestal bearings *f*, Fig. 163, are first unbolted and a rope sling passed around the ends of his rocker shaft. Since the shaft is lifted, the entire valve rocker mechanism for this particular cylinder is removed. While this appears to be a more difficult task than the removal of part of a rocker arm as in Fig. 157, actually it consumes no more time and, if the plant possesses a traveling crane hoist, is no more difficult.

Valve Cam Clearances.—The roller clearances to be maintained are as follows:

Admission valve cam clearance	0.039 in.
Exhaust valve cam clearance	0.036 in.
Fuel valve cam clearance.	0.002 in.
Starting valve cam clearance .	0.036 in.

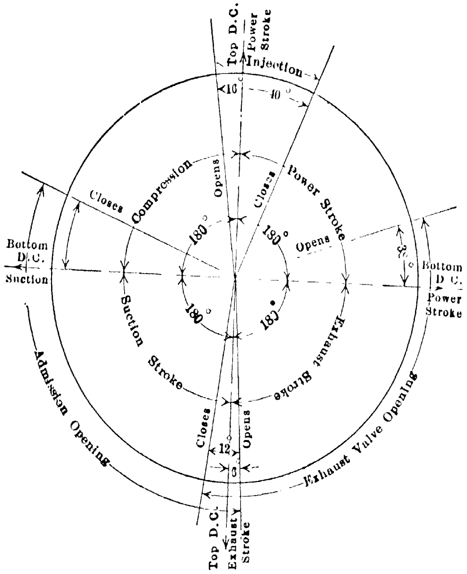


FIG. 165.—Valve timing McIntosh & Seymour 500 hp. engine, 164 r.p.m. four cylinders $18\frac{7}{8}$ dia. × $28\frac{3}{4}$ stroke.

Figure 165 gives the valve timing which is correct for the 500-hp. engine at 164 r.p.m., and this equally applies to all sizes of the slow-speed engine. The manufacturers set the admission and exhaust valves on the erecting floor and do not recommend any change. The helical gears are marked to enable the erector to assemble the valve mechanism. Nevertheless, after a few years of service the gear teeth and the cams wear to such an extent that the valves do not function as originally set. It then becomes necessary to retime the valves along the settings indicated above.

The rocker shaft is provided with small grease cups. Usually the lubrication so obtained is insufficient, and the cups should be replaced by sight-feed oil cups.

Lombard Diesel Valves.—The admission and exhaust valves of the Lombard engine are interchangeable and appear in Fig. 71*A*. Each valve is provided with a cage and, as indicated in Fig. 71*A*, the valves are placed at a 60-deg. angle in the cylinder head. In this way the head is made slightly concave. In addition larger valves are possible than with the usual arrangement of flat heads. The valves are also somewhat different as to the spring arrangement. These are not placed entirely within the cage but extend beyond the cage top, being held by a washer arrangement on the valve stem.

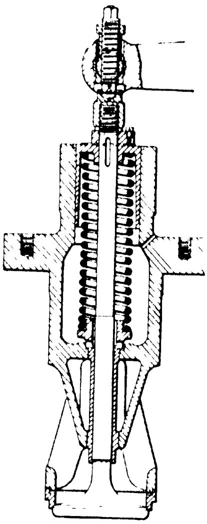


Fig. 166.—Admission valve, Fulton-Diesel.

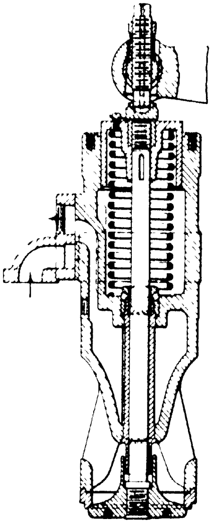


Fig. 167.—Fulton exhaust valve.

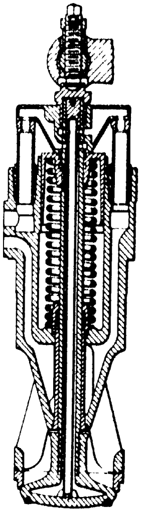


Fig. 168.—Valve of large Fulton engine.

The camshaft lies along the cylinder top and is provided with a cam for each valve rocker arm. The rocker end bears upon a hardened lock-nut or cap on the valve stem.

Adjustment of clearance between roller and valve is thorough alteration in the setting of the lock-nuts mentioned above. The usual roller clearances are

Admission valve roller clearance	0.02 in.
Exhaust valve roller clearance	0.02 in.
Fuel valve roller clearance	0.002 in.
Air starting valve roller clearance	0.02 in.

Fulton Iron Works Valves.—The admission valve, Fig. 166, and the exhaust valve, Fig. 167, of the standard Fulton engines follow customary lines. The exhaust valve, which is of cast iron, is threaded onto the valve stem, the latter being riveted over. The extension of the stem guide together with the form of the valve prevents burning of the stem.

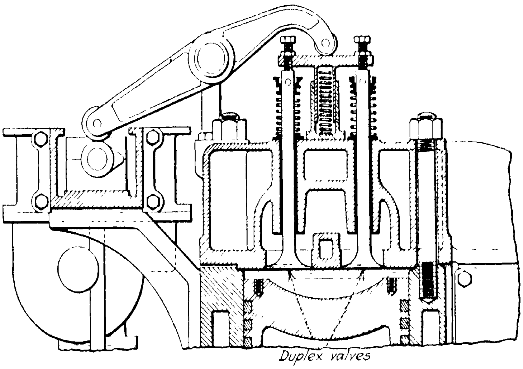


FIG. 169.—Winton valve gear.

The largest units have exhaust valves as well as the valve cage that are water-cooled. To allow the water to enter the valve stem a telescopic arrangement is placed on the valve stem as shown in Fig. 168.

The roller clearances are:

Admission valve roller clearance	0.038 in.
Exhaust valve roller clearance	0.036 in.
Fuel valve roller clearance	0.002 in.

The valve timing differs but little from those already given for other engines.

Winton Diesel Valves.—The Winton engine has two admission and two exhaust valves per cylinder. These are arranged about the center of the head where the fuel valve is located. The two exhaust valves are operated by one cam which bears on a yoke as illustrated in Figs. 169 and 170. The upper ends of the valve stems are threaded and pass through the yoke. Adjustment of clearance is had by placing washers between the yoke and valve stem.

For a high-speed Diesel the use of two exhaust and admission valves enables better scavenging and charging to be obtained. Besides, it is reasonable to expect a good cooling effect in the head and little danger of head fracture. The drawback, if any, lies in the necessity of keeping twice the usual number of valves in condition.

The camshaft lies along the side of the cylinder head, being driven by a vertical shaft geared to the engine shaft.

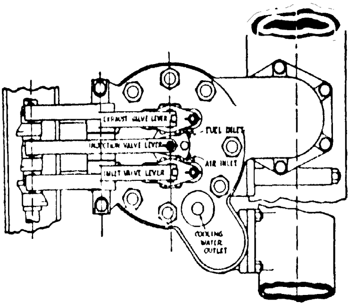


FIG. 170. Top view, Winton valve gear.

In the Marine engine the camshaft is slid longitudinally to bring the reverse cams under the valve-rocker rollers. Figure 170 shows the cam arrangement.

Snow Diesel Engine Valves.—Figure 171 outlines the admission valve while Fig. 172 is the exhaust valve assembly of the Snow horizontal engine. The valve stems are identical, while the valve bodies differ in that the exhaust valve body is extended along the stem, forming a gas deflector to avoid the burning of the stem. The valves have one feature that makes them rather peculiar—the valve seats are flat surfaces. The advantage of the flat valve lies in its freedom from any wedging action, with the consequent grooving which, at times, occurs with the bevel seat, and in the increased valve opening over the bevel-seated valve for any given lift. There is, however, an objection to the flat valve which an occasional engineer voices. It is much more difficult to keep gas-tight, largely due to the accumulation of small carbon particles on the face. The valves are easier to regrind, and consequently any deposits can be quickly removed.

The valve stem is equipped with labyrinth grooves, and it becomes necessary to clean these recesses of tar each time the valve is inspected. Since the side pressure from the rocker arm is against the stem, the stem must be copiously lubricated; this is obtained by a line from the mechanical oil pump on the engine frame. The cage is sealed by a ground joint at the inner end as well as by a gasket at the outside flange.

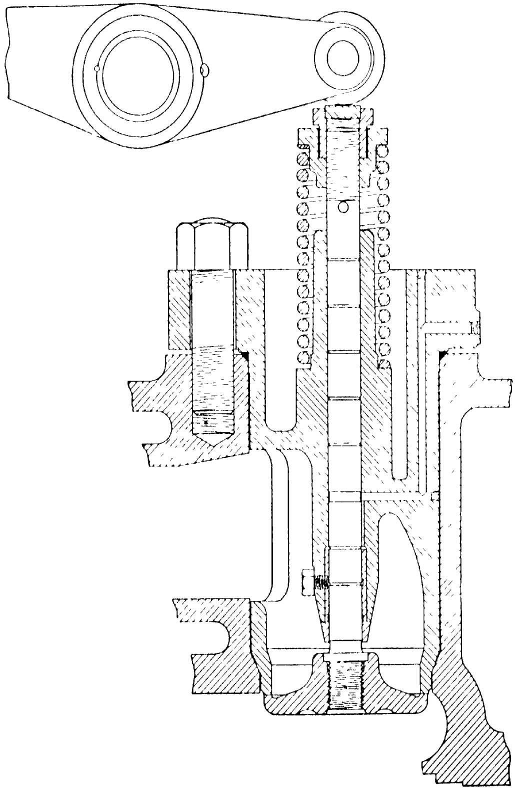


FIG. 171.—Snow admission valve.

Camshaft.—The camshaft, as will be noted in Fig. 173, is driven by bevel gears from the engine layshaft. The rockers are fulcrumed on a short shaft which is supported by the cylinder head. To remove a valve cage, the entire rocker assembly must be unbolted. This fulcrum shaft is provided with an eccentric bushing upon which are fitted the starting and fuel-valve rockers.

Cam Clearances. The proper roller clearances are as follows:

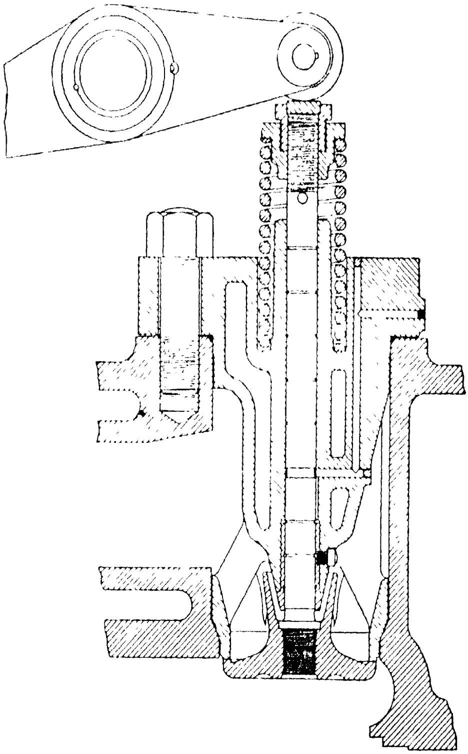


Fig. 172 Snow exhaust valve

Admission valve cam clearance	0.01 in.
Exhaust valve cam clearance	0.01 in.
Fuel valve cam clearance	0.004 in.
Air starting valve cam clearance	0.036 in.

The proper valve timing appears in Fig. 174. Any change of the entire cam timing can be obtained by shifting the gears one

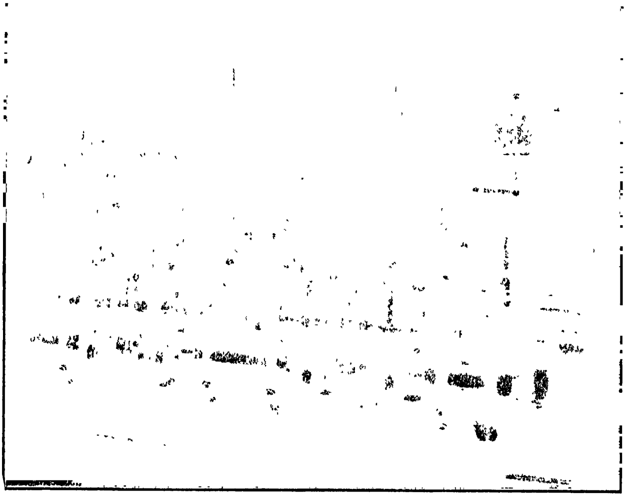


FIG. 173.—Snow camshaft and valve rockers.

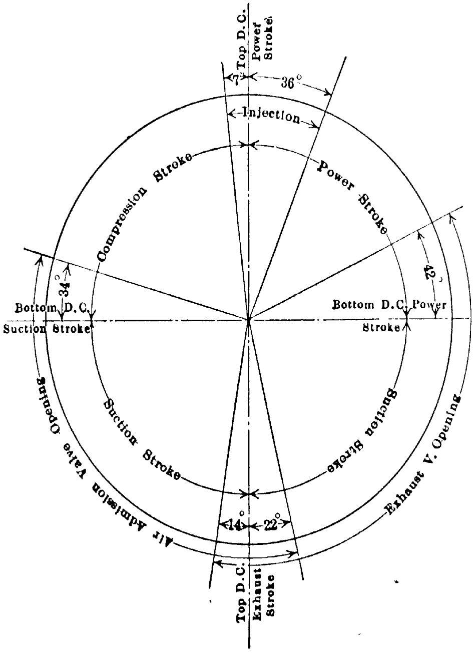


FIG. 174.—Snow valve timing.

or two teeth, while the alteration of an individual valve cam can be secured by shifting the cam nose.

Worthington Marine Diesel.—In Fig. 175 is shown the admission valve of this 1,750-hp. Marine Diesel. The cage only is water-cooled while both the exhaust valve and its cage is cooled by water.

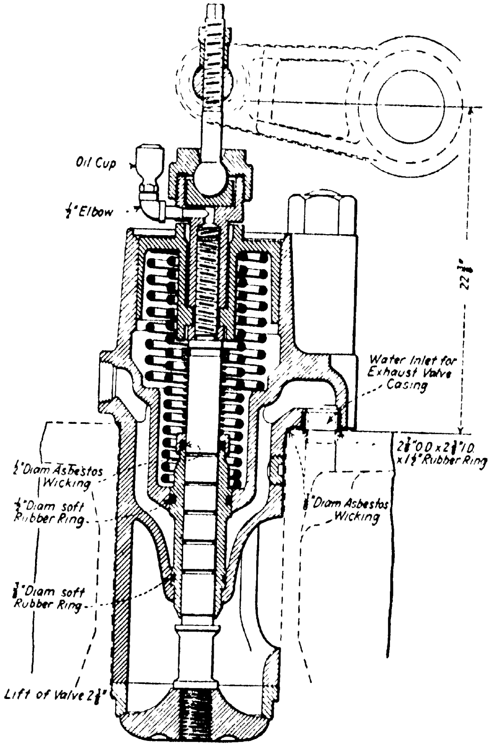


FIG. 175.—Worthington exhaust valve.

In addition to the main exhaust valve in the cylinder head an auxiliary spring loaded exhaust valve is placed in the cylinder barrel well toward the bottom end. The piston, as it nears the end of the power stroke, uncovers this valve. Since the spring holding the valve seated has but little strength, the pressure opens the valve, allowing the major part of the charge of hot gases to pass out through this valve. The main exhaust handles only a small part of the gases. On the compression stroke, the air has a very low pressure during the interval before the piston covers the auxiliary valve, consequently the valve remains closed. This feature should do much toward lessening the number of stoppages at sea to regrind valves.

The various valves are opened by rockers actuated by pushrods. The rods are seamless steel tubing with welded forged steel ends. The lower ends, which bear rollers, are linked to the reversing shaft, as is shown in Fig. 176. The roller clearances are from 0.030 to 0.040 in.

FIG. 176.—Worthington camshaft.

The reverse and control are obtained in a very simple and positive manner. An air cylinder is provided with a piston and rod. The rod at its upper end is fitted with a rack and a cam plate. This latter bears against rollers fastened to the camshaft. The rack engages a pinion keyed to the reverse shaft. When the handle of the reversing valve is thrown to "astern," the air enters the lower part of the air cylinder. This forces the piston rod upward. The rack revolves the reverse pinion, which rotates the reverse shaft. The links connected to the valve pushrod then pull the rollers out of contact with the cams. The cam plate on the reverse piston rod has moved upward during this period, and as the rollers clear the cams the curve on the cam plate engages the roller block and shifts the entire camshaft to the right, bringing the astern cams under the rollers. Further movement of the rack brings the pushrod back onto the astern cams. An oil cylinder, equipped with a bypass valve actuated by the reverse lever, effectually locks the reverse in position. The engine is now ready to run astern.

Two control levers are provided, thereby allowing three cylinders to be cut out when running at low speed. After the reversal has been accomplished, the control lever is moved inward. When the lever reaches its first position, the air valve is opened. As the engine comes up to speed under the influence of the air which is admitted into all the cylinders, the control lever is shifted farther inward; this cuts out the air and admits the fuel. The fuel pump is controlled by altering the duration of opening of the pumpsuction valves.

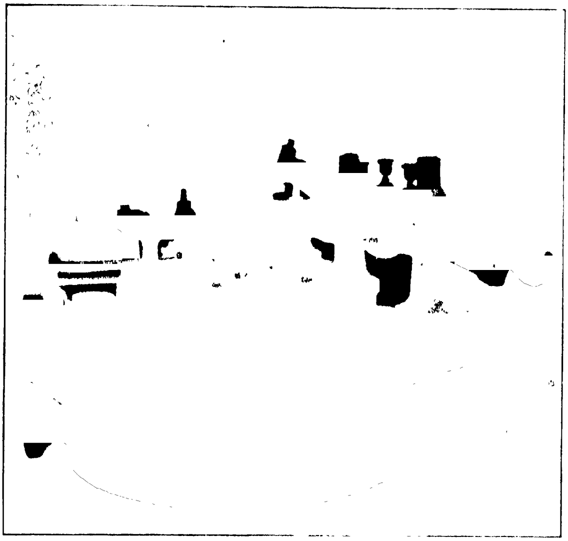


FIG. 177 Nordberg FD valve gear.

Snow Vertical Diesel.—The design of valve used on the Snow Vertical Diesel is shown in Fig. 111. The most noteworthy feature of this engine is the cooling of the valve cages by the same water that passes through the cylinder head. Thimbles lead the water from the head to bosses on the valve cages.

The valves are driven through rocker arms by the layshaft which runs along the cylinder head, see Fig. 75.

Nordberg Two-cycle Diesel Valves.—The large Nordberg Diesels are fitted with four scavenging valves in the head. These are arranged in pairs, and each pair is controlled by a rocker arm moved by the camshaft. The arrangement, as shown in Fig. 177. The clearance between rocker and cam varies from 0.03 to 0.04 in. depending upon the engine size.

The smaller engines use port scavenging, and so the head carries only the fuel and starting valves which are in one cage.

McEwen Diesel Valves.—One excellent feature of this horizontal engine is the provision of cages for both the exhaust and the admission valves. These cages and valves are mounted in the cylinder head in vertical positions, as appears in Fig. 70. Both valves are of cast iron with steel stems; the exhaust valve body is extended to form a hood about the lower part of the stem. The two valves and the admission valve cage are shown in Fig. 178; both valves have 45-deg. seats. To remove either cage, it is only necessary to unship the reach-rod pin and lift out the cage with the rocker attached.

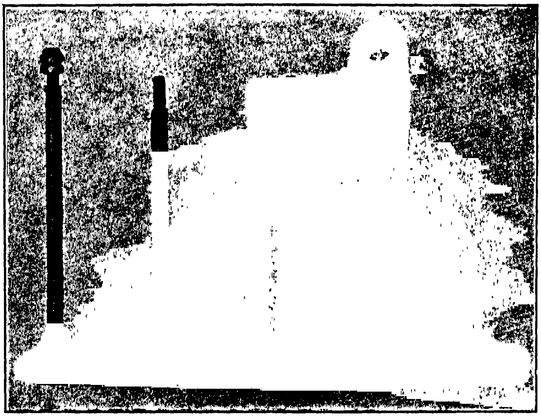


Fig. 178.—McEwen admission valve and cage.

The camshaft is driven by a helical gear fom the engine shaft. The valve rocker arrangement is outlined in Fig. 88. Each valve has its individual cam, the exhaust cam roller being provided with a shifting pin which relieves the compression on starting the engine.

Cam Clearances.—The valve cam clearances follow:

Admission valve cam clearance	0.005 in.
Exhaust valve cam clearance	0.03 to 1/32 in.
Injection valve cam clearance	0.001 in.
Starting valve cam clearance	0.03 to 1/32 in.

Valve Timing.—The timing diagram applicable to the engine appears in Fig. 179. These values can be followed by the operator with a feeling that they are correct.

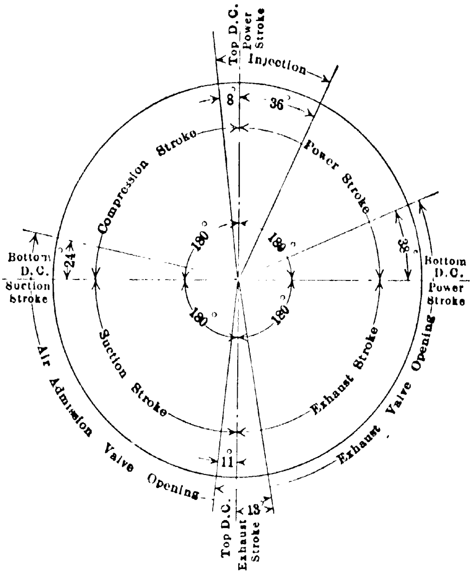


FIG. 179.—McEwen valve timing.

National Transit Diesel Valves.—This engine has the admission valve provided with a cage while the exhaust valve has a half-cage, which carries the valve spring and stem housing. The latter valve does not seat on the cage but instead a removable seat is set into the cylinder casting and held in place by machine screws, Fig. 180.

The valves differ in that the exhaust valve body is extended to form a guard around the stem. In removing the valves, the admission valve cage is first lifted off, and then the exhaust valve spring and cap are unshipped. The exhaust valve can then be raised through the admission valve cavity. If it is desired to remove the exhaust cage, a rope can be passed through the valve stem housing and tied; slipping the other end through the admission cage cavity and hooking it to a hoist, the exhaust cage may be lowered to the floor or lifted into place as the case may be. The false exhaust seat is difficult to remove when worn as the screw threads tend to burn and corrode fast. Vigorous hammering with a long bar will frequently loosen the screws; kerosene should always be poured on the seat since this will assist in cutting away the iron rust from around the screw threads.

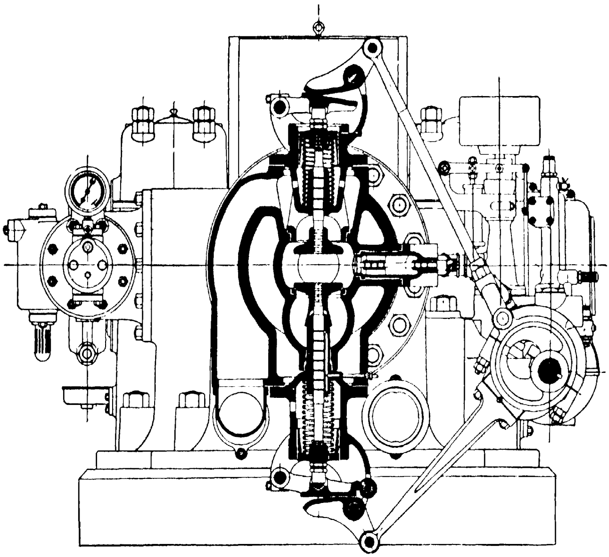


FIG 180.—National Transit valves and rockers

Valve Cam Mechanism.—Both the valves are actuated by wiper type levers. These levers are controlled by a single eccentric as shown in Fig. 180. Since the contact between the flat valve lever and the rocking levers is a rolling one, the clearances are not of such serious moment. The correct timing of the valves can be secured by either shifting the eccentric, or altering the length of the valve stems through the agency of the valve lock-nut caps, or by adjusting the length of the valve reach-rods. The latter is the proper method for securing any minor timing adjustments. The valve timing of this make of engine appears in Fig.

181. It is apparent from Fig. 13 that the air-starting and fuel-injection cams are distinct from the valve eccentric.

National Transit Diesel, 1918 Model Valves.—The later engines produced by the National Transit Co. have a valve and camshaft design radically different from the one discussed above. The valves are placed horizontally in the cylinder head, and both are provided with cages. This entire assembly appears in Fig.

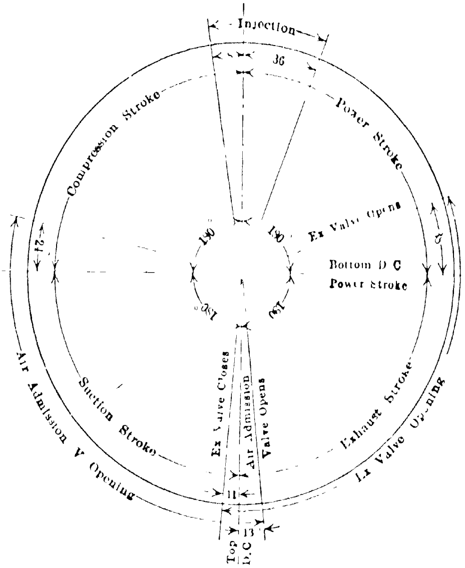


FIG. 181.—National Transit Diesel valve timing

182. The end of the valve stem has an adjustable head which is in contact with the valve rocker, and a collar against which the valve spring rests. No spring cap or other bearing is provided for the support of the outer end of the stem. However, the cage sleeve is long and gives ample support.

Camshaft Layout.—The rocker arms fulcrum on a short shaft which is carried on bearings bolted to the head; this shaft is loose in the bearings and is held by the starting lever shown. The rocker rollers rest on the cams which are keyed to the short camshaft; this shaft is driven by the layshaft through bevel gears.

The advantage of this design lies in the better cylinder head that is permissible with horizontal valves.

Valve Timing.—The timing of the valves is the same as applies to the former design and appears in Fig. 181.

Allis-Chalmers Diesel Valves and Rockers.—The valve arrangement of this engine, Fig. 183, is quite similar to the one just discussed in Fig. 180. The exhaust valve, however, has no removable seat since it rests directly on the cylinder-head casting. Use is made of a single eccentric for both valve actuating rods. In the two-cylinder engines a system of links allows the one camshaft to control the exhaust and admission valves of both cylinders. The valve rockers have a wiping action and consequently open and close the valve, while the rocker is moving at a maximum speed, without shock or noise. The noiseless valve action is especially noticeable to one accustomed to the slam-

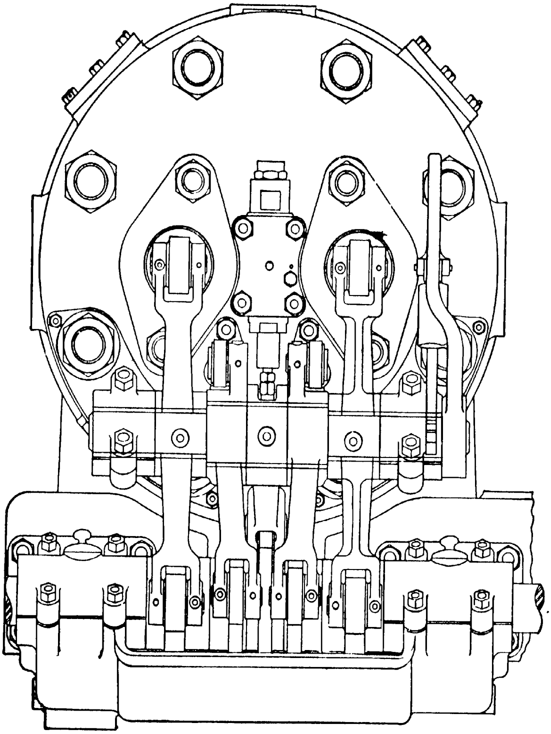


Fig. 182.—National Transit 1918 design valve rocker assembly.

ming noise of the usual cam-controlled Diesel valves. Since this form of valve actuating levers is used, no clearance is required between the lever and valve cap when the eccentric is in its extreme low position, beyond that necessary for the oil film, which need not be more than 0.001 inch.

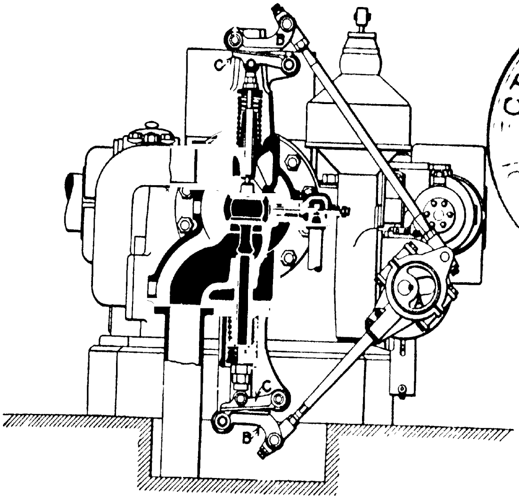


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Fig. 183.—Allis-Chalmers valves and rockers.

In the later models of the Allis-Chalmer's Diesel where more than one working cylinder is used, a different valve gear is employed. A short eccentric shaft runs along the cylinder heads, being driven by the layshaft through beveled gears. Eccentrics on this shaft drive the valve rocker arms, the valve rocker is turned 90 deg. from the position shown in Fig. 183.

Standard Fuel Oil Engine.—This horizontal engine, being of the two-stroke-cycle type, possesses no air admission or exhaust valves. The piston performs the duty of both valves, uncovering air scavenging ports and exhaust ports at the forward end of its stroke. The piston action, along with the functioning of the scavenging air valve, should properly fall within the scope of this discussion of engine valves. In Chapter VIII a few words were devoted to the scavenging piston which acts as a compressor to supply the scavenging air to the power cylinder, although the means of this accomplishment were not mentioned. Figure 77 represents the engine cross-section, showing both the power piston and the scavenging piston, which is bolted to the former. Figure 184 is a view of the injection air compressor. The low-pressure air piston is of an hour-glass shape and acts as a valve in directing the proper flow of the scavenging air. The pipe *a* is the air suction line open to the atmosphere. The air, flowing in along the pipe *a*, passes through the port *l*, allowing this air to enter the scavenging air cylinder *H*, shown in Fig. 77*A*. As the engine turns over this air is compressed, and the continued movement of the plug piston *g*, Fig. 184, uncovers the port *d*, which permits the air to flow out through the pipe d_1 into a low-pressure air receiver, not shown. This air is at a pressure of 8 to 10 lb. gage. After the fuel charge in the power cylinder fires and the piston moves outward, the exhaust ports *K* are uncovered. At the same time the scavenging air ports *e* are also opened. During this interval the plug piston *g* has moved, shutting off the connection between the scavenging cylinder and the air-receiver line *d*, and has placed *d* in communication with the scavenging air ports *e*. The air, stored in the receiver, rushes through the ports *e*, scavenging the power cylinder of the exhaust gases. It is apparent that the successful functioning of the entire air-scavenging system depends on the condition of the air-compressor piston or plug valve *g*.

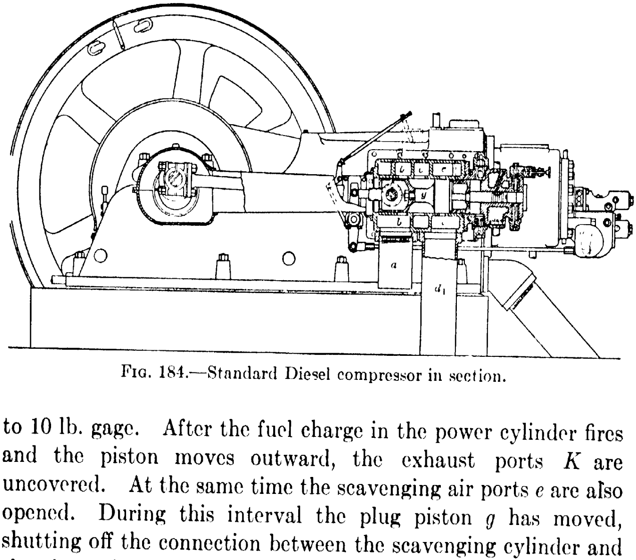


FIG. 184.—Standard Diesel compressor in section.

Standard Fuel Oil Vertical Diesel Valves.—The vertical Standard Fuel Oil engine has a somewhat simplified arrangement of the scavenging device. The shipped piston, Fig. 70*A*, delivers the scavenging air through flap valves into a chamber running along the cylinders. The piston, on uncovering the scavenging air ports, deflects the air currents upward along the cylinder, forcing the burnt gases out of the cylinder. The scavenging is excellent, and the engine develops full power even at high speeds where it might be expected that the scavenging would be poor.

De La Vergne FD Diesel Valves.—This horizontal Diesel, of which only a few have been built, has the valves located horizontally in the head. They are driven by rockers which are actuated by a camshaft bolted to the front of the cylinder head, as appears in Fig. 23.

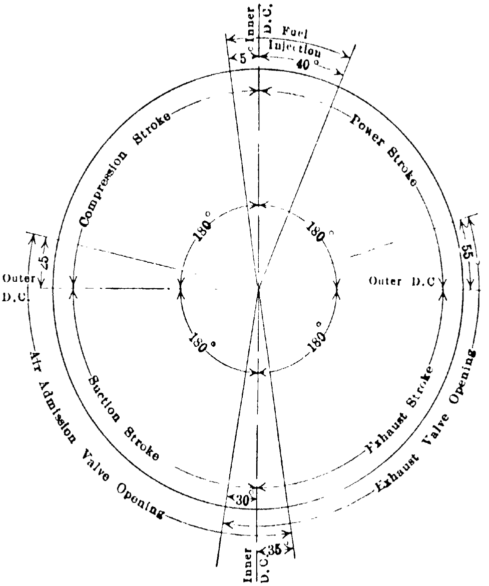


FIG. 185.—De La Vergne FD Diesel valve timing

Valve Timing.—Figure 185 gives the valve timing of this Diesel. Individual units may vary slightly from these values.

McIntosh & Seymour Trunk Piston Marine Diesel Valves and Valve Cages.—The admission and exhaust valve cages of the trunk type engines are made of cast iron and do not have separate valve seats. The valves proper are also of cast iron with steel stems cast in, and are guided in the lower part of the valve cage and do not have an upper guide as is used in the stationary Diesel for the guiding of the valve.

Rocker Arms.—The various rocker arms are operated from the cams by means of vertical push-rods, including the air starter and fuel valve rockers, Fig. 186.

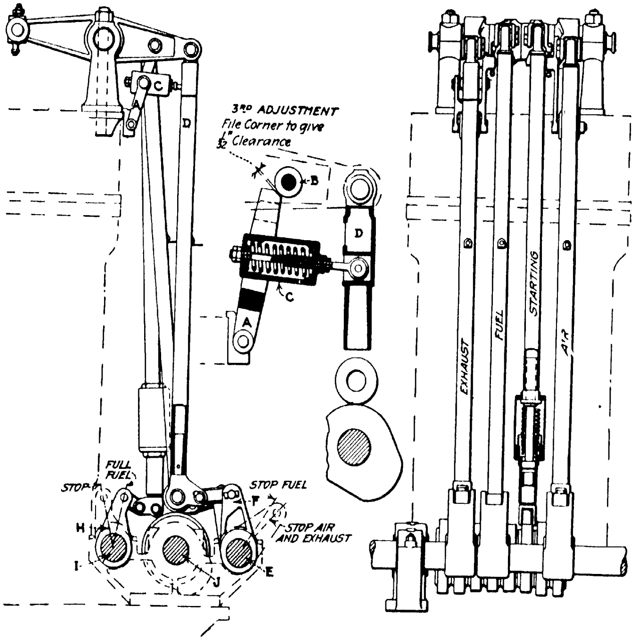


FIG. 186.—McIntosh & Seymore trunk-piston marine Diesel valve arrangement.

Camshaft.—The camshaft is driven by a set of spur gears from the crankshaft at the after end and runs at half the engine speed. A double set of cams is arranged side by side in the following order: exhaust, fuel, starting, and admission. They are keyed to the shaft and bolted together. Cams and rollers are made of cast iron and are chilled at the running edges.

Reversing and Operating Gear.—This gear is located on the forward end of the engine. Two turns of the reversing wheel shift the camshaft in a horizontal direction for either ahead or astern rotation. This mechanism is designed as follows: The

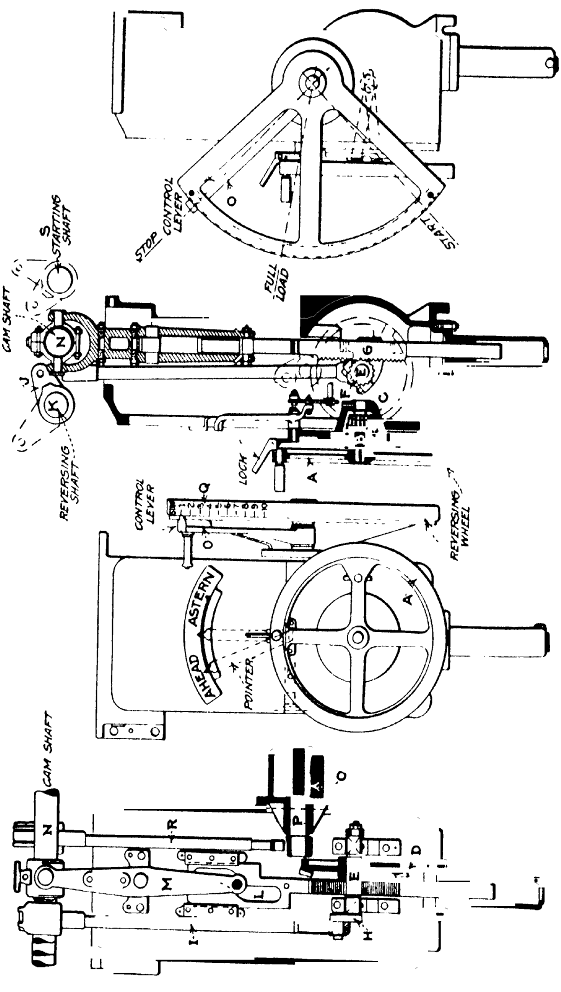


Fig. 187.—McIntosh & Seymour reversing apparatus.

handwheel turns a shaft to which a bevel gear is keyed, Fig. 187, which in turn rotates a large bevel gear *D* mounted on another shaft, at right angles to the first one, carrying a pinion which operates a rack *L* up and down. This rack has an extension which is provided with a slot in which a roller moves, which roller is fastened to a rocker arm *M* pivoted around a fixed point. The other end of the lever is forked and its arms fit in a groove in a sleeve fastened to the camshaft, thus moving this shaft in either direction at will. The rollers for exhaust, fuel, and admission are moved away from the respective cams by the rods *I* and *R* at the same time the camshaft is shifted longitudinally. Starting is performed by moving the lever *O*, which controls fuel and starting, all the way downward, in which event the starting air roller is brought in contact with its respective cam, admitting air to the cylinders. As soon as the engine has turned over a few times on air, the control handle *O* is moved upward again, bringing the fuel admission at the full-load position. It will be understood that the starting air was automatically cut off again as soon as the control lever reached the "full-load" mark on the segment.

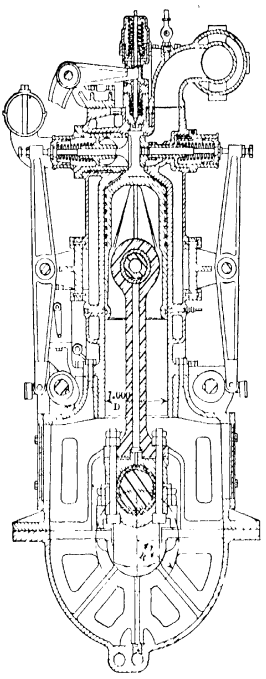


Fig. 188.—Small Nelseco marine Diesel.

When the engine is run on fuel, the reversing gear and the fuel regulations are interlocked, and only when the control lever is on STOP does it become possible to turn the reversing wheel as the notch is disengaged from the bevel gear rim. The control lever cannot be moved unless the engine is fully reversed either way.

Nelseco Four-cycle Marine Diesel Valves and Valve Gear.—Figure 188 is a cross-section of one of the small Nelseco marine Diesels. The admission valve is provided with a cage, and the exhaust valve seats on a removable seat in the combustion chamber walls. The air and fuel valves are actuated through levers by the starboard camshaft while the exhaust camshaft is on port side. The two camshafts are driven by a 2 to 1 reduction gear from the engine shaft.

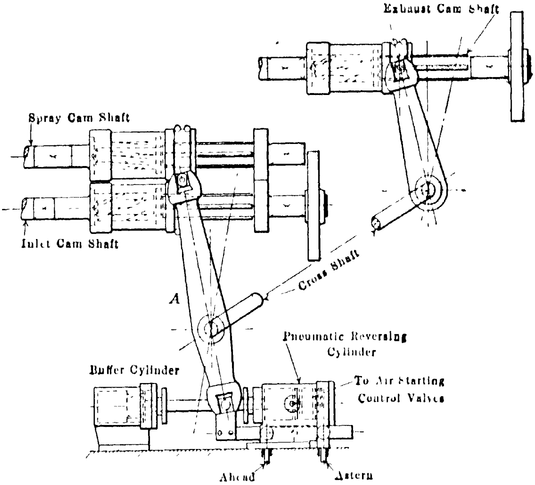


Fig. 189 — Nelseco reversing mechanism

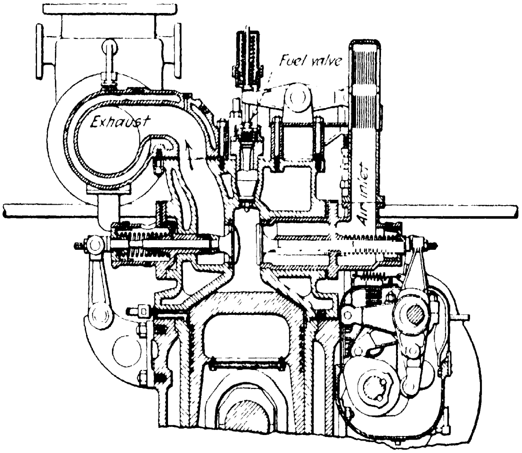


Fig. 190.—Valve gear of large Nelseco engine.

Reversing Gear.—The majority of these Diesels are equipped with a gear for reversing, though some have a direct reverse arrangement shown diagrammatically in Fig. 189. The camshafts, of which there are three, have spiral keyways cut in them. Each shaft has a collar with a spiral key which is shifted by a pneumatic reversing cylinder. A movement of the rocker *A* throws the collars to one side, thus turning the camshafts through an angle that will set the valve cams correct for reverse running. Due to the varying angle through which the three sets of cams must move, the spiral keyways do not have the same pitch, thus securing proper shifting of the cams.

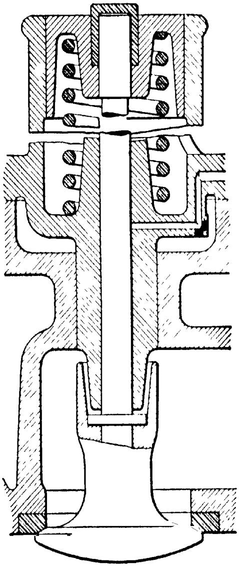


FIG. 190*A*.—Nelseco exhaust valve.

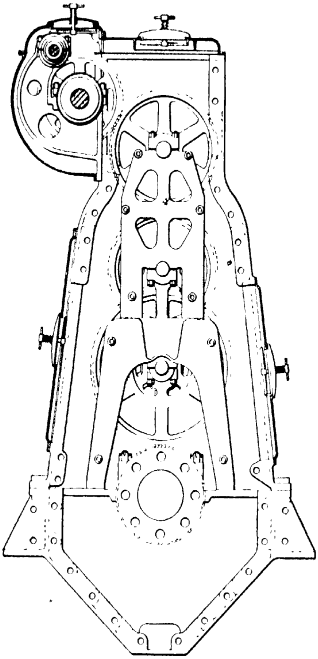


FIG. 190*B*.—Camshaft gear train.

The larger Nelseco Diesels have a valve gearing somewhat different from Fig. 188. The valve gearing of the 600 hp. stationary engine is shown in Fig. 190. It will be noticed that the valve rockers are driven by a cam shaft which runs alongside of the engine, close to the cylinder heads. A reachrod, not shown, runs from one rocker arm across the cylinder head and is connected to the exhaust valve arm. This design eliminates the long rocker arms used on the smaller engines as well as reduces the number of layshafts. The camshaft is driven from the crankshaft through a train of gears, Fig. 190*B*.

Valve Cages.—As would appear from the foregoing discussion, all the manufacturers use cages on the admission valves while the practice is not universal as regards the exhaust valves. There can be no doubt that the cage design is superior and is of advantage to the operator. When a valve seat leaks, a spare valve and cage can be used to replace the defective parts. No matter how thrifty the operator must be as to the capital invested in replacement parts, it is the height of economy to carry one extra exhaust valve with their cages.

Water-cooled Cages and Valves.—Due to the severe temperatures of the exhaust gases as they pass through the exhaust valve cage, water cooling of the valve stem housing is well-nigh imperative in engines above 12-in. cylinder bore. All manufacturers provide for this stem cooling even though a valve cage is not included in the valve design. It is clear that the circulation of water must be positive if the sticking of the stem is to be avoided. Where the water carries any considerable percentage of mineral salts, there is a decided tendency toward the scaling-up of the cooling space, which is, at best, of small volume. Consequently, each time the exhaust cage is removed the water-cooling cavity should be filled with a 10 per cent muriatic acid solution and afterward thoroughly flushed out with a stream of water under at least 20 lb. pressure. Since the engine has already extracted the available portion of the heat of the gases, there is no loss occasioned by a cool valve cage; the cage-cooling-water discharge should be kept below 120 deg. F. as a low temperature improves the valve action. Even at this temperature of the discharge the valve stem is actually much hotter, and the lubricating oil often forms a gumming deposit that makes the valve motion erratic. A little kerosene should be injected along the stem after each shut-down.

Water-cooled Valves.—On some of the larger engines water-cooling of valves, especially the exhaust valve, is essential to the successful functioning of the valve. The water is customarily introduced at the side of the stem and flows down into the body, where it enters a central tube which carries it to a connection at the end of the valve stem. With both water-cooled valves and water-cooled cages the water must be circulated before the engine starts firing. If this matter is deferred until the engine has warmed up, the sudden chilling of the parts by the cold water will invariably produce a fracture.

Pitting and Corrosion of Valve Seats.—Leaky exhaust valves may be due to any of the following causes: Cutting effect of the highly heated gases passing at high velocity between valve and seat; carbon or dirt from the fuel becoming caught between valve and seat and hammered into the metal; corrosive effect of acid in the fuel; warping of valve disk, causing uneven seating; valve stem striking in guide, preventing the valve from seating firmly.

Cleaning Valves.—Every three months each valve and cage should be removed for inspection and cleaning. A regular schedule can be followed whereby the valves of one cylinder are removed every three weeks; in a four-cylinder engine this gives a three week inspection, while in a three-cylinder engine an inspection every three months is secured. When the engine is a single- or double-cylinder unit, the schedule should be arranged so that the tri-monthly examination is obtained. At these inspections the valve and cage should be completely disassembled and thoroughly cleaned with kerosene, washing off with gasolene. If a valve and cage is kept on hand, the old one can be cleaned at leisure. This spare set is very essential where an engine operates almost continuously. The time required for removal and replacement of cage and valve should not exceed 30 min. when the engine-room force is well organized.

Grinding Valves.—If the valve seat becomes rough, allowing the compression to escape, it must be reground. Where the valve seats in a cage the unit is disassembled. The valve and dashpot or spring cap, as it is more popularly termed, is replaced in the cage with a light spring resting between the valve body and the stem housing, along the lines of Fig. 191*A*. The cage is inverted, placed on some form of support, and the valve pin wrench is then set in position. The spring raises the valve off the cage until a slight downward pressure is exerted by the man doing the regrinding. A mixture of emery flour and oil or emery flour and vaseline should be coated over the valve face and the valve rotated back and forth. The valve should, in no case, be completely revolved; the rotation or movement should cover a trifle more than a quarter circle. After rotating a few minutes, the valve should be moved another 90 deg. and the rotation renewed. This grinds every portion of the valve face to conform to the entire seat of the cage. As the operator grinds the valve, he should release the downward pressure on the valve, allowing it to rise from the seat. This serves to distribute the emery paste over the entire face. If this is not observed, the paste forms at the edges only, causing the valve seat to be ground concave.

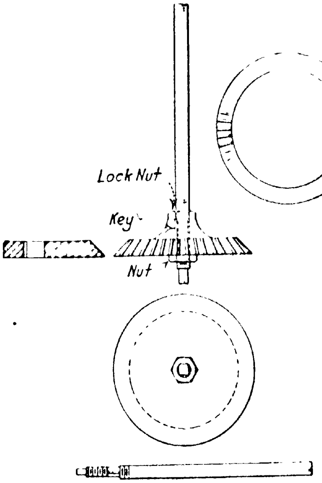


Fig 191.—Valve seat reamer.

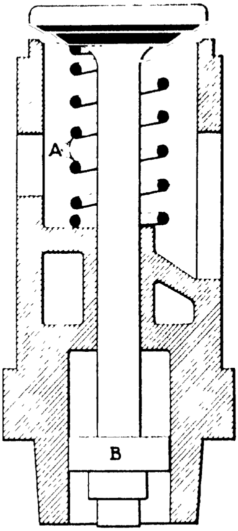


Fig. 191*A*.—Regrinding valves.

The operator need not secure a ground seat over the entire valve face. A narrow contact $\frac{1}{16}$-in. wide is ample; in fact, a line contact of $\frac{1}{32}$-in. width is as serviceable a gas seal as is a wider space. After the seat is sufficiently ground, the emery paste can be removed. The valve should then be rotated without any paste between the two surfaces; this metal to metal grinding or rubbing will make the area of contact as smooth as a mirror and prolongs the tightness of the valve.

When the valve becomes so pitted or grooved that grinding will not be of any avail, the cage and valve faces must have a light cut taken off their surfaces. The valve can be centered in a lathe and a finishing tool used, making the cut as light as possible. Exhaust valves in the small sizes are made of alloy steel, but for the larger sizes it is becoming more and more customary to make the valve of cast iron, with chilled surfaces, and fit it on a steel stem. In such cases recourse is had to the grinding wheel, as the metal is too hard to be cut by a lathe tool or reamer.

To reface the cage a refacing machine, similar to a rose reamer, is necessary. This reamer must have a stem resting in the valve-stem housing to hold the reamer square with the valve stem. Most manufacturers are in a position to supply this machine, although any machine shop can build one like the sketch in Fig. 191. The cutting head should be of tool steel while the stem can be either of tool steel or machine steel. The latter is preferable since the cutting head can be hardened and the machine-steel stem turned to bring the cutter concentric.

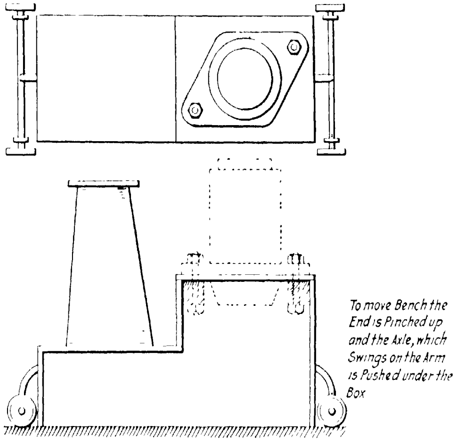


FIG. 192.—Bench for grinding and reaming valve cages.

Few plants possess any stand to hold the cage other than a wooden box. The box is a poor accessory since it is almost impossible to seat the irregular-formed cage on it with any feeling of security. A wooden grinding stand, similar to Fig. 192 can be made during odd hours and certainly repays all labor spent on it. The opening in the top should conform to the shape of the cage while the two stud-bolts should fit into the regular stud-bolt holes in the cage. With such an apparatus an engineer can sit down while grinding, thus lessening as much as possible the labor involved in this operation.

A valve grinding arrangement shown in Fig. 193 is used in some of the stations of Sinclair Pipe Line Co. The stand is made out of 1-in. pipe and fittings. Two short nipples are screwed into tees at the top of the stands and these nipples engage the bolt holes of the valve cage. A short lever is clamped to the end of the valve stem by lock-nuts and a long link hooks into the end of this lever and has its other end fastened to the plunger guide of the reciprocating pump. In this way a rotating motion is given the valve and the engineer occasionally lifts the valve by means of the lever shown at the bottom of the stand.

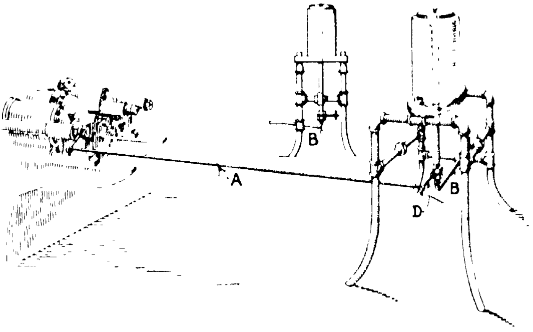


FIG 193.—Valve grinding apparatus.

On new engines it will nearly always be found that both the exhaust and air-inlet valves will take a permanent set after a short time in use and must be taken out for regrinding, after which they are less liable to deformation. In the case where exhaust and air-inlet valves are duplicates, it is advisable after a few weeks running to interchange them. This is known as "seasoning" the valves. In some engines, especially the larger sizes, the blow-torch action of the fuel spray causes the exhaust-valve disk to become much hotter on the side adjacent to the fuel-injection valve. This uneven heating may cause serious warping of the valve. If the valve construction is such that the valve can be rotated by means of a wrench on the upper part of the stem while the engine is running, the operating routine should call for giving the valves a turn at least once every hour. This will tend to prevent permanent warping, due to the uneven heating.

Sticking of the exhaust-valve stem in its guide may be caused by too much clearance permitting gummy oil and carbon to work up between stem and guide, as in the case of the spray valve. This action is accelerated when very heavy fuel oil is being used, by the fact that the heavy, unburned residue that may be carried in the exhaust gases in a finely divided state strikes against the exhaust-valve stem and sticks. This action occurs when the valve is pushed downward to the open position, thus exposing a part of the stem that travels up into the guide when the valve closes.

When a valve is removed for overhaul, the stem should be carefully cleaned, as described in the case of the spray valve. Both the valve and its seat should be examined for fine heat cracks that may develop into leak paths. It is good practice to run kerosene through the valve-stem lubricator occasionally while the engine is in operation. This tends to clean out gummy oil that may collect in the guide.

The treatment of air-inlet valves should be the same as for the exhaust valves, but it will be found that they will not require regrinding as often. Due to the cold air passing through the inlet valve during the suction stroke, it never gets as hot as the exhaust valve, and it has none of the dirt and unburned oil in the exhaust gases to handle.

Valve Timing.—In timing valves the first step is the establishment on the flywheel of the points of dead-center of the cranks. The simplest method of marking the dead-centers is to use a steel trammel having both ends pointed. A steel block, with a counter punch mark on the surface, can be inset into the foundation, being held by lead or cement grouting, Fig. 194. To establish her flywheel position, when the crank is on upper dead-center, one of the valve cages can be removed and the distance from the surface of the cylinder head to the piston, when the piston is approximately at upper dead-center, can be measured and a trammel mark made on the flywheel rim. The wheel is then turned on over past dead-center until the piston is again the same distance from the cylinder head. A second mark is placed on the flywheel rim; the bisection of the distance between the two trammel marks gives the exact dead-center for the piston in question. A second but not so exact a method is the use of a spirit level on the crank throws; this, of course, is impossible if the throws are not machined accurately on all four sides. The center mark should be placed on the flywheel rim and stamped with the cylinder number, as example 2TC, indicating top dead-center of No. 2 cylinder. The same procedure is followed on all the cylinders for both top and bottom dead-centers. These center positions being determined, the next step is the checking of the exhaust valves. The engine should be turned over until the piston of the cylinder in question is about 50 deg. from bottom dead-center. A steel tape line can be used to measure the distance on the flywheel rim from the bottom dead-center line to the point of correct exhaust-valve opening. Since the timing given is in degrees, the value must be transformed into inches on the flywheel circle. Presuming the wheel is 8 ft. in diameter, the circumference being 302 in., each degree represents, very closely, $\frac{5}{6}$-in. If the proper opening of the exhaust is 42 deg. ahead of bottom dead-center, a distance of 35 in. is set off ahead of the dead-center mark. This is spotted and stamped E2O; that is, exhaust opening for No. 2 cylinder. The flywheel should then be turned slowly until the trammel cuts this E2O mark. The exhaust cam rocker should be firmly in contact both with the cam and with the valve stem. To be doubly certain, the adjusting screw on the rocker should be backed off and brought up again until the operator can feel the resistance of the valve spring against the screw. In case, while revolving the wheel, the mark E2O is passed, the valve should not be checked by bringing the wheel back to the mark. Instead, the wheel should be brought back past the mark at least 12 in. and then again turned until

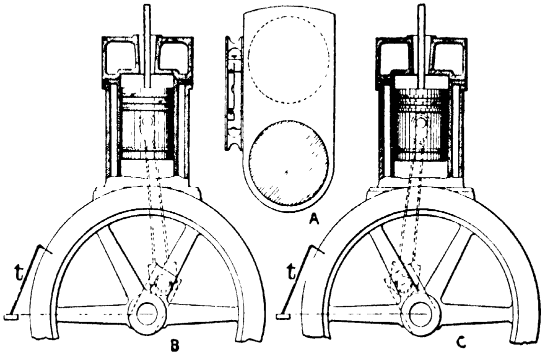


FIG. 194.—Locating crank centers.

the trammel cuts the mark. This removes all effect of any back lash that might exist in the cam gears. The same process is followed in checking the exhaust closing position. After this is accomplished, the exhaust valve settings of the other cylinders are checked, the flywheel being properly marked for each position. The admission valves should next be gone over in proper sequence, and all points should be indicated on the wheel for future reference.

In event any of the valves fail to check up correctly, the operator is confronted with the question as to the method of changing the setting. If the discrepancy is only a few inches, the clearance between the cam and the valve rocker can be changed, bringing the setting back to the stated values. If the valve opens vastly early, or late, the only recourse is the shifting of the cams by the use of an offset key. This condition is encountered in old engines only. In these engines quite often the entire valve layout is timed late. This is attributable to the wear in the cam gear teeth and can be partially corrected by advancing the cam gear one or two teeth.

The average operator, until he is very familiar with engine timing, does well to time the exhaust and admission valves of one cylinder before proceeding to any of the valves of the other cylinders. There is, in this way, little danger of becoming confused as to the proper stroke upon which the valve should function. The trained operator customarily checks the valves as they come in sequence. For example, the exhaust opening of one cylinder will be set, then the admission closing of a second cylinder will be checked. This reduces the time occupied in going over the valves by at least 75 per cent. The checking and setting of the fuel valves will be taken up in the chapter on fuel valves.

It is possible to place a dial upon the end of the cam shaft as shown in Fig. 215. Many builders of marine Diesels fit a similar dial before the engine leaves the factory.

Leaky Valves.—A leaky admission valve is readily detected by placing one's ear close to the intake nipple or screen. If the leak is from a scored seat, a whistling noise will be heard; this whistling increases in volume as the scoring becomes more pronounced.

A leaky exhaust valve is more difficult to detect. The best method is to place the engine in that position where both the admission and exhaust valves are closed; that is, at the beginning of the power stroke. The injection air, line valve should be "cracked," allowing a small amount of air to blow into the cylinder through the fuel-injection valve. If the exhaust leaks, the air can be heard flowing through the valve. This same procedure will also locate admission-valve leaks. While the engine is running, a smoky exhaust and a decrease in power are often due to a leaky exhaust valve.

CHAPTER XIII

FUEL INJECTION VALVES

TYPES. ACTION. ADJUSTMENT. REPAIRS

Injection Design of the Original Diesel Engine.—Dr. Diesel in his first patent application, made no mention of any mechanism for the forcible injection of the fuel. The fuel contemplated was coal dust, and this was to be deposited in a pocket of the rotary valve *V*, which, in revolving, dropped the charge into the cylinder. A schematic outline of this fuel-conveying apparatus is shown in Fig. 195.

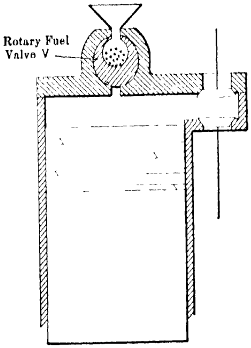


FIG. 195.—Fuel feeding arrangement, Dr. Diesel's original patent.

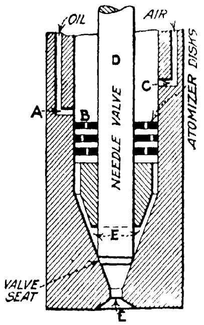


FIG. 196.—Fuel valve, closed type.

This design was never followed in an actual engine since it was early seen that it would not deliver the fuel charge properly. Furthermore, the plan of using coal dust was abandoned in favor of oil. Dr. Diesel in conjunction with the M. A. N. Co. of Germany, adopted the idea of employing a blast of air to break up the oil charge and deliver it into the cylinder.

Injection Action.—For the benefit of those unacquainted with the functioning of the fuel-injection valve the following brief explanation is given. The charge of fuel oil is pumped into a receptacle called the fuel valve by some form of pump. An air compressor delivers a charge of pure air at about 900 lb. gage to this injection device. At the proper moment in the engine's cycle the needle valve of the injection device, or fuel valve, is opened, connecting the cylinder with the fuel supply. The high-pressure air then rushes into the cylinder, carrying the oil charge with it. This fuel, as it is forced through a device called the atomizer, which is located in the valve housing or cage, is broken up into fine fog-like particles that will ignite when intermingled with the cylinder charge of air, which is at a high temperature. Figure 196 shows such an elementary form of fuel valve where *D* is the valve stem; *A*, the oil line; *C*, the air line; *B*, the atomizer; and *E*, the opening into the cylinder.

It is evident that the fuel valve has two main functions. First, it must allow the oil charge to be introduced into the engine cylinder or combustion chamber at the proper time. Second, it atomizes or breaks up the stream of oil in such a thorough manner as to occasion ignition. The cylinder temperature is high; the value corresponding to 550 lb. compression pressure should be at least 1,000 deg. F., when the engine is cold; after warming up, the temperature would be well above 1,400 deg. F. If the oil charge was injected in a solid mass into this highly heated air, the oil would vaporize and burn but a very slow rate. The air would not be thoroughly mixed with the oil, and the combustion would occur only on the surface of the oil mass, in exactly the same manner as a pool of oil burns when ignited. The objection to this latter method lies in the slow rate of combustion and in the loss of fuel which escapes as unburned gas vapors. If the oil charge is separated into many minute particles, more surface area is presented to the air. If this oil separation process is to be successful, the fuel valve must be provided with some means whereby the oil is broken into particles and mixed with the injection air before the oil enters the engine cylinder.

If the oil spray was to be finely divided and pumped into the cylinder without the air injection the energy of the oil particles would not be enough to enable the oil drops to penetrate the air mass. The injection air supplies this energy.

The idea of using a charge of high-pressure air was taken up in the first experimental Diesels and has been retained on all "true Diesels" to the present day. The air charge in the fuel or spray valve, being at a pressure 300 or 400 lb., higher than the cylinder pressure, provides an excellent means of forcing the oil through the valve passages into the cylinder as soon as the needle valve opens. The expansive power of the air as it drops in pressure from 900 to 500 lb. on entering the cylinder increases in volume and tears the oil into minute particles. A well-designed valve will break up the oil so finely that, if the engineer holds his hand in front of the fuel nozzle, the oil will not even wet the hand but will float as a mist.

By reason of the chilling effect of the air in expanding through the valve from its high pressure to the existing cylinder pressure, a drop of 50 to 100 deg. may be experienced by the cylinder air charge. This necessitates a slightly higher compression pressure and temperature than is required by the "solid-injection" oil engine discussed in later chapters. The chilling effect is of consequence only at the moment the needle valve opens and starts the flow of air and oil. After the oil ignites, any further chilling is of little importance, for the cylinder temperature upon ignition of the oil immediately rises to from 2,000 to 2,700 deg. F., and the chilling effect of the air injection is of little detriment. It is evident then, that the elimination of chilling when the spray valve opens is desirable. This may be and is generally accomplished by so designing the valve that a small amount of oil enters the cylinder ahead of the air. This oil upon ignition raises the cylinder temperature high enough to avoid any bad effect of the air chilling. At the same time the amount of oil so entering must be small, for not being as finely broken up as the balance it would, if of any great amount, causes soot or carbon deposits.

If the oil is hurled into the cylinder in a mass, its combustion, even though imperfect, would cause the pressure to rise to an abnormal value. The Diesel engine is presumed to burn the oil at constant pressure, at or but slightly higher than that of compression. To accomplish this the oil must be introduced at a comparatively slow rate. The two main offices the valve performs are that of "braking," or offering a resistance to the oil flow, and that of thoroughly atomizing the charge. The latter would not be difficult of achievement if there were no other considerations entering into the problem. The same conditions prevail as to the "braking" action. If this "braking" effect is obtained by the imposition of a series of improperly designed baffle plates or disks, the air pressure required to force the oil through these resistances may become so great as to make the method impractical. The desirable fuel valve is one that thoroughly atomizes the fuel and exercises a control, or a "braking" action, over the fuel charge without any great loss of air pressure through the valve.

The design of the orifice between the needle valve and cylinder has a decided influence upon the conbustion within the cylinder. The combustion space in all Diesel engines is limited in volume since the compression ratio is large. If this volume (about 1/15 of the stroke volume) be in the shape of a cylinder between a flat piston top and a flat cylinder head, it is difficult for the oil to mix with the entire air charge. Instead, without proper nozzle design, the oil will be localized in the central portion above the center of the piston crown. The oil will be imperfectly burned due to the limited amount of air in contact with the oil spray. Carbon deposits will appear on the center of the piston head and lead to overheating and fracture of the piston head unless the piston cooling be good. If the oil is introduced along the axis of the cylinder head, as is the usual custom, it must be spread by some means. In view of the short length of the combustion space of a flat-headed engine the engineer will perceive that this spreading is by no means easily accomplished. It would, on first thought, seem that a series of outlets would distribute the oil better than a central orifice, but results of the two different types of openings apparently differ but little. It has been found on certain engines that the use of a multi-holed orifice or flame plate has improved conditions. With a flaring central orifice the oil seems to spread about as well in most cases.

Classes of Atomizers or Fuel Valves.—The many fuel valves employed on the various makes of Diesel engines fall into two classes—the closed-nozzle and the open-nozzle valves. The former is the one adopted on the pioneer Diesels and is found on all vertical engines of the present day; in fact, this is necessary for structural reasons. The open nozzle, known in Europe as the Lietzenmayer nozzle, is largely used on the horizontal engines, both of the domestic and foreign manufacture.

Open-nozzle Fuel Valve.—In the design of this nozzle, or fuel valve, the needle valve controls the flow of air to the atomizer tip, this needle valve being opened at the proper time by a cam-actuated rocker. Between the valve and the cylinder is interposed a small cavity, or enlargement of the passage to the cylinder; into this cavity is deposited the fuel charge. Since the fuel is pumped during the suction or exhaust stroke of the piston, the fuel pump works against only a few pounds pipe-resistance pressure. This enables the governor to be very sensitive in action since the reaction of the pump is at a minimum. At the proper time the needle valve is opened and the air from the compressor, as it sweeps along the passage to the cylinder, picks up the oil charge and carries it into the cylinder. The swirling of the air serves to break up the mass of the fuel; this is further increased by perforated disks or other devices at the nozzle tip. Figure 197 outlines this class of injection valve. It would appear that the duration of the fuel injection would be in exact ratio to the amount of oil deposited in the atomizer and that the first particles of oil entering the cylinder should be as completely nebulized as are the last few oil droplets. In actual practice the air, when it strikes the fuel charge, produces slugs of oil that enter the cylinder in an unatomized condition. Where perforated disks intercept the slugs, as in some designs, the entire charge is satisfactorily broken up. The only real objection the operator can well offer is the rapid carbonization of the fuel passage and atomizer disks; this can be attributed to the absorption of heat from the combustion chamber since the temperature in the nozzle tip fairly approaches that existing in the cylinder on the compression stroke. This necessitates a more frequent cleaning of the atomizer than is required by a closed type of nozzle.

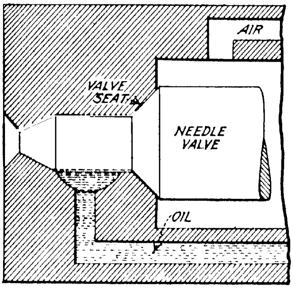


FIG. 197.—Open nozzle fuel valve, elementary form.

The open-type valve is of advantage where dirty oil is burned. Since nothing but pure air passes the needle valve, the scoring of the valve seat, so prevalent with the closed-nozzle fuel valve, is entirely absent. Consequently, the air valve requires less than half as much regrinding as does the closed-type needle valve.

Closed-nozzle Fuel Valve.—The earlier Diesels employed the closed nozzle, and it was exclusively used on all Diesels until ten or so years ago. Figure 196 outlines the basic principle of this type of nozzle. The fuel valve has a body in which a cavity is formed, enclosing the atomization device. The fuel needle valve is seated below this device and is actuated by a cam-controlled lever. The interior of the fuel valve is in connection with the air line through the passage and is at all times under an air pressure of 900 lb. per square inch or more. The fuel pump forces the oil charge through the line *A*, the oil settling around the valve stem at *E* above the valve seat. When the needle valve opens, the compression pressure of the engine is around 500-550 lb., while the air pressure in the valve body is about 900 lb. This pressure difference results in a rapid flow of the air into the cylinder. The air charge forces the oil along with it, and, in passing through the tortuous passages of the atomizer disks, the oil is completely nebulized.

This form of fuel valve has the advantage of depositing the oil in a receptacle entirely isolated from the influence of the hot compressed air in the cylinder. Furthermore part of the oil, being immediately around the valve tip, enters the cylinder ahead of the air and ignites, even though it is not thoroughly atomized. This primary ignition provides a flame to fire the remainder of the oil, which enters the cylinder at a somewhat low temperature due to the expansion of the air charge at the valve tip. Unfortunately, with many closed-nozzle designs, an entirely too great a percentage of the fuel enters the cylinder ahead of the air; in some it appears that all the oil is forced ahead of the air. If the disks are designed with perforations of small diameter to enable the air to mix with the oil, the "braking" or resistance of the atomizer is increased since the disks are entirely filled with oil at full load. This compels the employment of a higher injection pressure, or the time interval of fuel injection is prolonged. On low loads, with small perforations, the fuel charge is of small weight and does not flow down around the valve seat. The consequence is that the first part of the injection consists of air only, which, in expanding, chills the nozzle tip and housing. This delays the combustion, producing a smoky exhaust.

Hardly any two Diesel manufacturers follow the same design in the fuel valve, although there is some degree of similarity in a few designs, especially of the open-nozzle type.

American Diesel Co.—One of the pioneer fuel valves was that of the American Diesel engine. This injection device was contained in a cast-iron housing, which was bolted to the side of the cylinder head, as shown in Fig. 198.

The needle valve is opened by the bracket lever shown, which, in turn, receives its motion from a reach-rod and cam, the latter being mounted on the engine camshaft. The closure of the needle valve is accomplished, as is the universal practice, by a spring, which has been compressed by the opening of the valve.

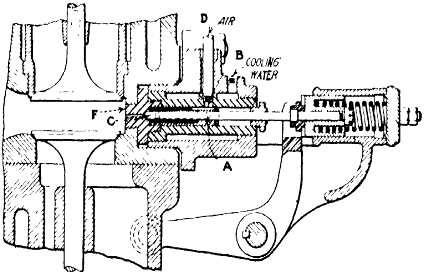


FIG 198.—American Diesel fuel valve assembly.

The housing of the fuel valve carries the needle valve, which is surrounded by the atomizer body. This atomizer is similar to Fig. 199. The fuel enters the fuel-valve housing or cage through the passage *A* and flows along the recess between the atomizer and the valve body until it reaches the front or valve end of the atomizer. The recess is enlarged, at this point, to include the entire circumference of the atomizer. From this recess or ring a series of minute passages lead to the needle valve immediately above the valve seat. The injection air enters the housing at *D* and flows around the atomizer, behind the oil; it also fills the space about the needle valve stem. When the valve *C* opens, the flow of air around the stem results in a drop in pressure from the back of the valve toward the valve tip. The air pressure

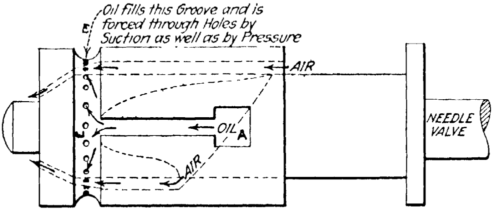
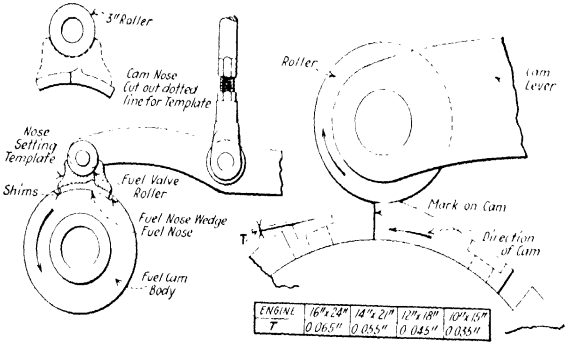

Oil fills this Groove and is forced through Holes by Suction as well as by Pressure
E
AIR
OIL
A
AIR
NEEDLE VALVE

FIG. 199.—One design of American Diesel atomizer.

behind the oil charge, on account of the larger volume, remains practically constant and at a higher value than the pressure around the valve seat. This pressure difference forces the oil through the small ports E into the stream of high-velocity air rushing through the needle-valve opening. The air thoroughly separates the oil into minute particles; this breaking-up process is further intensified by the flow of the oil and air through the openings in the nozzle-tip plate F between the valve seat and the cylinder. The disadvantage of the valve lies in the chilling effect of the air charge. The design allows a considerable part of the air to enter the cylinder ahead of the oil; since there is no resistance from an oil body present, this air attains a high velocity and, in expanding, lowers the temperature of the nozzle tip as well as exerts a similar effect upon the air which has been compressed in the cylinder.



ENGINE	16"x24"	14"x21"	12"x18"	10"x15"
T	0 065"	0 055"	0 045"	0 035"

FIG. 200.—Setting American fuel valves.

The fuel valve is equipped with a by-pass valve of the screw-needle type; this valve is for the purpose of draining the valve cage of any oil. In stopping the engine, the by-pass valve is opened and the oil flows through the valve back to the storage tank. The lack of fuel then causes the engine to stop. In starting the engine, the fuel pump is operated, by a hand crank, until oil appears at the by-pass overflow, indicating the fuel line is free from air. The by-pass valve is then closed.

Timing of Fuel Valve.—The fuel valve of the American Diesel engine should be timed as outlined in Fig. 156A. The fuel-valve cam has an adjustable tool-steel nose, the position of which can be varied to secure proper valve opening.

To set the fuel-cam nose in its correct position, the crank of the cylinder in question is placed on upper dead center. If the cam nose has straight sides, as in the sketch at the right, Fig. 200, the nose is shifted until a thickness gage placed in line with the mark on the nose touches the cam. The thickness of the gage for the several engine sizes are shown.

The earliest engines had a cam with a curved nose. To set this properly a template is cut out of sheet iron and a mark made on the template in line with the mark on the nose. A spare cam roller is placed on the template with its center on the gage line and the roller outline scribed on the template. After cutting the roller pattern out, the template is set between the roller on the rocker arm and the fuel cam nose on the engine. The nose is shifted until the template fits both roller and nose. This is the correct position of the nose when the crank is on upper dead-center.

Adjustment and Repair. By-pass Valve.—The American Diesels in service have been operating a number of years, and consequently frequent repairs are to be expected. One defect that will early develop is the leaking of the by-pass valve. The valve is of the needle type, and the corrosion of the point allows oil to seep at all times. This can be eliminated by redressing the point of the valve, followed by a regrinding.

Needle Valve.—The needle valve has a rounded end which seats on a bevel surface of the atomizer shell. To regrind the valve, the end should be rounded and all rough spots removed. The valve can then be inserted into the atomizer, being coated with extra-fine emery paste and ground to a proper contact with the seat. Bronze needles are often used in order to avoid the corrosion that occurs when an oil having much sulphur is used along with a steel stem. Since the bronze stems are rather expensive many plants use a cold-rolled steel valve stem with the lower part of bronze screwed into the steel stem.

Valve Stem Stuffing-box.—The gland of the valve stem stuffing-box is rather weak, while the stuffing-box is very shallow. The air will blow out along the stem, and in tightening the gland stud-nuts the ears of the gland will invariably bend. If the stuffing-box were deeper, enough packing could be inserted to withstand the air pressure. The only available remedy is the substitution of heavier glands, enabling more pressure to be exerted on the nuts. The packing used is vulcanbestos and wears away rapidly. In renewing the packing, the rings should be soaked in oil for at least twenty-four hours. When in use, the valve stem must be lubricated frequently to maintain the packing in a pliable condition. The fuel-valve disk, which lies between the valve seat and the cylinder, carbonizes very heavily when a low-gravity fuel is burned. A lye solution will loosen this deposit.

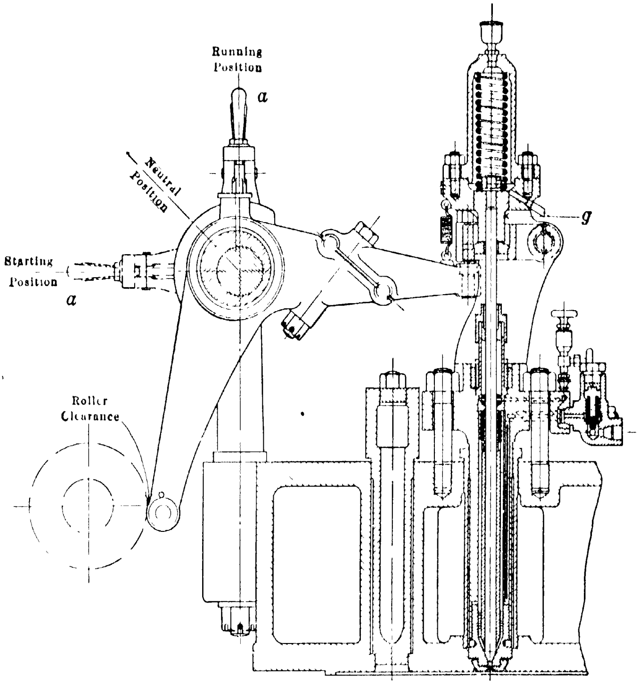


Fig. 201.—Fuel valve arrangement Busch-Sulzer Type B Diesel.

Busch-Sulzer Type B Diesel.—The fuel valve of this Diesel, with its rocker mechanism, appears in Fig. 201, while Fig. 202 gives a view of the lower part of the same valve. The valve consists of a cast-iron body with an extension which carries the spring, a needle valve and an atomization device. The body or cage rests in a bushing, which is pressed into the cylinder head, and is held by two studs. The needle valve *a* is enclosed along the lower part, which is in the cage, by a bushing or barrel *b*. This bushing rests, at its base, on the atomizer cone and is prevented from lifting by a coil spring at the top. As is seen in Fig. 202 there is a space, between this bushing and the cage wall, which serves as the air and fuel cavity. The fuel charge flows down the passage *A* and enters the fuel cavity above the atomizer disks. The disks are several plates containing small perforations and are so placed as to stagger these holes. The oil, by its own weight, is forced to pass through these openings and fills the intricacies between the plates as well as the space around the needle-valve seat. The air enters the cavity *d* above the oil level. When the needle valve is opened by the valve rocker, the air, which is at a pressure much higher than that existing in the engine cylinder, forces the oil charge into the combustion space. The oil, as it is broken up, passes from disk to disk, and is mixed with the air which is flowing toward the cylinder. The emulsion is further increased as the air and oil issues through the single small opening in the atomizer tip *f*. This continues as long as the needle valve is open; this valve is closed at the proper time by the fuel cam, and the small amount of fuel still left in the atomizer flows down around the needle-valve seat. This insures a small charge of oil which will enter the cylinder at the next valve opening ahead of the air and produce ignition. With the usual fuel oil the fuel charge is able to pass through the disks under the influence of its own weight. When the valve opens, the oil is forced into the cylinder in front of the air. Since this oil is fairly light, it ignites even though poorly broken up. A heavy oil behaves quite differently. Its viscosity is such that the fuel does not readily flow through the disks but rests above them. The air must force the oil through the atomizer and, in so doing, thoroughly nebulize the charge. This type, then, has the advantage of offering a mixing and atomizing effect in inverse ratio to the gravity of the fuel; the actual

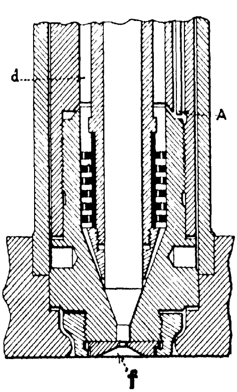


Fig. 202.—Busch-Sulzer Type B fuel valve.

degree of combustion in the cylinder is fairly constant, regardless of the nature of the oil.

Adjustments—Fuel Passage.—The fuel passage in the body of the valve cage is equipped with an aluminum rod which stands vertically; the rod moves with the vibration of the engine, thereby keeping the fuel line free from stoppage due to dirty or thick oil. Even with this precaution, the fuel line does collect dirt and should be flushed out with kerosene on the occasion of removal of the valve and cage.

Nozzle Tip and Disks.—The fuel cage tip has a single central perforation which gradually fills up with tarry deposits, as do also the atomizer rings or disks. These can be cleaned with kerosene or lye water.

Needle Valve.—The needle valve is of steel and will corrode when the fuel oil contains any acid or sulphur. If the corrosion is slight, the valve should be polished with emery paste and, if possible, burnished with a cloth buffing wheel. The buffing of the stem seems to retard the rate of corrosion. The valve seat is at an angle of 30 deg. with the axis of the needle valve. This angle, which gives an angle of 60 deg. to the valve end, is such that there is a wedging action each time the valve seats. This serves to seal the valve effectually but causes scores on the bearing surface in event any grit has settled on the valve.

Stuffing-box.—The stuffing-box has a screw gland and is best packed with lead or babbitt shavings or strings. This packing conforms to the valve sufficiently to prevent the loss of air and does not become hard as does all-composition packing.

Valve Spring.—The valve spring is of a length that allows the needle valve to be reground before the initial compression of the spring is lost. An iron washer placed above the spring will give the effect of a lengthened stem. However, the valve cannot be used after it is worn short enough to cause the spring plunger *g* to drop below its housing.

Fuel Line Check Valve.—The fuel line has a ball check valve immediately at the valve cage. This ball, in time, wears, allowing the high-pressure air to back through the oil line to the pump. This causes the line to become air bound.

Fuel Valve Rocker Arm.—The rocker arm is of two-piece construction and fulcrums on an eccentric bushing, which carries the air-starting rocker. The valve end of the rocker has an adjusting screw that can be manipulated to provide the required clearance between the roller and the fuel cam.

Fuel Cam.—The fuel cam is fitted with an adjustable steel nose. The nose is slotted and is held by two countersunk screws. This allows a considerable shifting of the nose. After the nose is set in position, it is locked by end shims. These shims are best made of wrought iron and should be hammered until the entire recess is filled; the surface should then be smoothed with a file to conform to the curvature of the cam.

Eccentric Rocker Bushing.—As has been mentioned, the fuel rocker arm is fulcrumed on an eccentric bushing, which also carries the starting rocker. On a four-cylinder engine the two inside cylinders are fitted with starting valves. When the engine is to be started, the eccentric lever *o* of these two cylinders is thrown into the starting position, Fig. 159. This revolves the eccentric bushing until the fuel rocker fails to engage its cam while the starting rocker comes into contact with its cam. The levers of the outside cylinders (1 and 4) are set to "neutral," which disengages the fuel rocker and cam. As soon as the air-line valve is opened, the two starting cylinders turn the engine over. After one or two revolutions of the flywheel, the levers of 1 and 4 are moved to the running position, admitting fuel to the fuel valves. When these cylinders start firing, the levers on No. 2 and No. 3 cylinders are moved from the starting to the running position, cutting out the air-valve mechanism and engaging the fuel rockers.

Servomotor.—On the larger Busch-Sulzer engines the timing of the fuel injection is altered at load changes. This is accomplished through the agency of a cylinder placed in front of the engine, containing a spring and piston. This cylinder is in communication with the low-pressure cylinder of the air compressor. The air-compressor suction is provided with a damper arrangement actuated by the engine governor. On low loads the air to the compressor is throttled. This results in a lower discharge pressure in the low-pressure cylinder; this, in turn, lowers the pressure existing in the servomotor. The spring then forces the piston downward. The piston rod moves a system of levers that actuates an auxiliary roller which is linked to the fuel valve rocker. A diagrammatical layout of the servomotor appears in Fig. 203. It will be noted that as the air pressure in the servomotor becomes lower, due to a lighter load, the auxiliary roller *a* moves upward, thereby allowing the cam nose to strike it slightly earlier in the engine cycle. The auxiliary roller is in contact with the fuel-rocker roller *b* at all times, being held by the links *c*. The auxiliary roller is set in such a position that if the injection opening is early, as on low load, this roller remains in contact with the cam nose for a smaller interval. Consequently, on full load the period of injection begins later and extends over a greater crank angle than it does on low loads. This is shown in Fig. 203 where the dotted lines indicate the roller positions on low load and the full lines are the full-load positions.

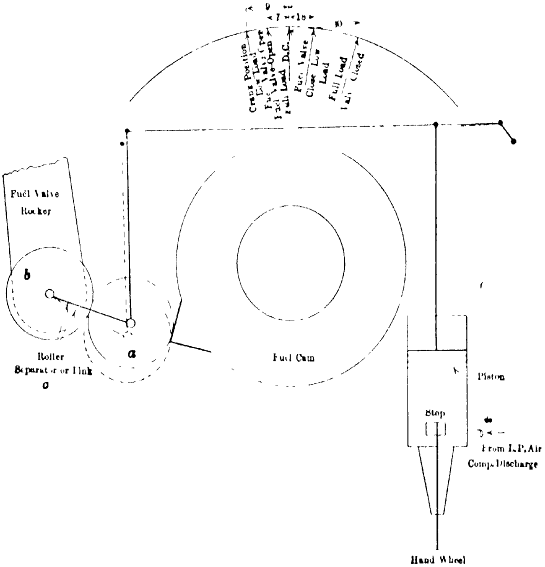


FIG. 203.—Method of altering fuel timing by servomotor.

On starting the engine, the handwheel of the servomotor is raised to the half-load position while the air from the air compressor to the servomotor is cut off. This latter action is for the purpose of preventing the servomotor from moving the injection roller to the full-load position. Frequently, when the engine is using heavy oil, it is necessary to adjust the servomotor handwheel to provide for earlier admission than the air pressure would give, or vice versa with heavy oil on light loads.

Fuel Valve By-pass.—Each fuel line has a by-pass, or relief valve, mounted on a block at the camshaft cover. Before the engine is turned over under air pressure, these valves should be opened and left in this position until a solid stream of oil issues from each valve, thus indicating that all air in the pump or oil lines has escaped. When running, frequently one or more cylinders skip in firing, or fail to fire at all; this, in most cases, is due to an air-bound fuel line.

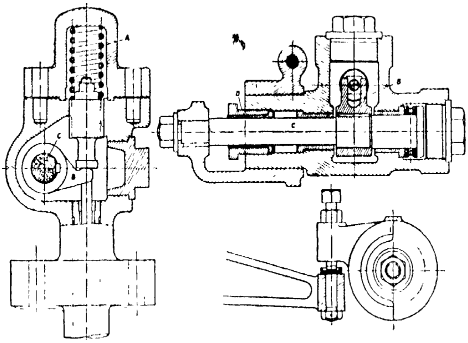


Fig. 204.—Busch-Sulzer fuel valve.

Busch-Sulzer Type C. Fuel Valve.—The fuel and air starting valves are placed in one cage. This requires the use of a single opening in the cylinder head, thereby reducing the danger of fracture.

The fuel valve spray disks, etc. are designed along the same, lines as the Type B. The needle valve is lifted by the rocker mechanism shown in Fig. 204 which shows the marine-type of connections. In the stationary engine the rocker is replaced by a link which is swung from the cam-roller lever, this lever in turn being suspended from the control shaft. Adjustment of clearance is obtained by means of the adjusting screw at the end of the valve rocker and on the stationary engines by adjusting the length of the link.

The air enters the valve casting at the lug and passes down the space between the sleeve and the needle, passing out at the lower end to the disks. The oil passes down a drilled passage in the casting and enters the cavity around the sleeve above the disks.

McIntosh & Seymour Diesel.—The fuel valve of this engine, appearing in Figs. 205 and 206, is contained in a cast-iron housing or cage. The valve, the needle end of which has a 60-deg. included angle, seats on the cage itself. The upper end of the cage carries the spring housing. This end of the valve stem is screwed and locked into a dashpot, against which the spring bears. The valve is opened by the movement of the rocker arm shown. The atomizer is the detail of the valve that differentiates it from other American fuel valves. This is a gun-metal barrel, the diameters of which are of irregular dimensions. The interior is bored taper, fitting the valve stem at the upper end, while a number of passages connect this taper bore with the outside surface. In operation of charge of oil is forced, by the fuel pump, through the passage *O* and surrounds the atomizer at the space *J*, issuing through the serrated fins at *K* and *L* until the fuel reaches the level *P*, Fig. 207. The injection air fills the cavity above the atomizer, and, when the needle valve opens, this air flows through the ports *B* and along the valve stem into the cylinder. The air pressure on the surface of the oil at *J* remains constant while the velocity of the air current along the

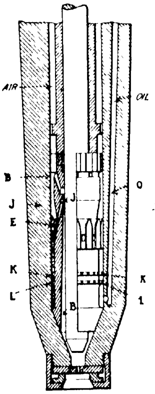


FIG. 205.—McIntosh & Seymour fuel valve.

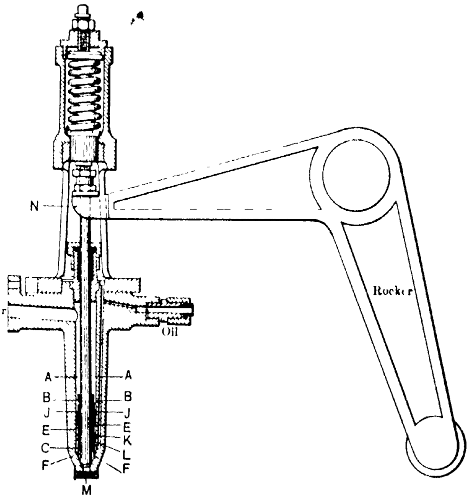


FIG. 206.—McIntosh & Seymour fuel valve assembly.

valve stem reduces the pressure at the inner end of the oil ports *E*. The consequence is a flow of oil through the oil ports *E* under the influence of the unbalanced air pressure. This oil, as it enters the stream of high-velocity air, is broken up and thoroughly atomized by the time it reaches the cylinder through the atomizer cap *M*. The flow of oil continues until the oil level falls below the ports *E*. If the fuel valve is properly timed, the valve should then close, preventing an excessive amount of air blowing into the cylinder. Although this excess air may actually increase the m.e.p. of the engine as figured from an indicator card, it represents a loss of power since it has been compressed to 900 lb. per square inch and is allowed to expand through the nozzle to 500 lb. per square inch without doing any work.

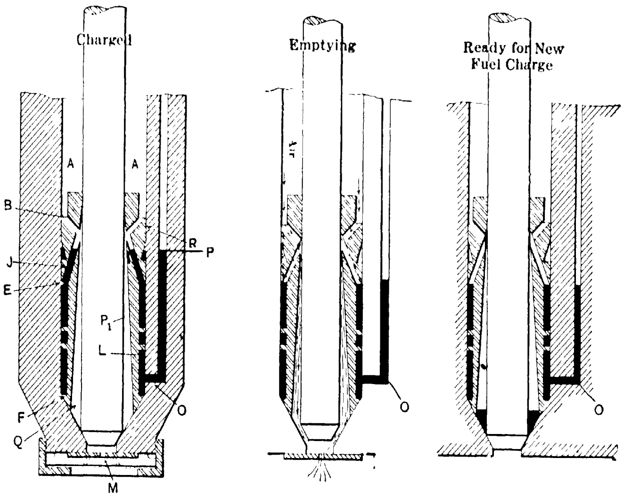


Fig. 207.—McIntosh & Seymour Hesselmann fuel valve action.

On full load the closure of the needle valve should trap a small amount of oil immediately above the valve seat. This oil, on the next valve opening, is blown in ahead of the air charge, providing an initial ignition to balance the chilling action of the expanding air charge. The oil, if it contains dirt or a tarry base, gums badly until the fuel chamber is filled, forcing the oil to deposit around the valve stem. When this occurs, the oil enters the cylinder in a slug. This is indicated by loss of power and a smoky exhaust. The remedy is, of course, the cleansing of the atomizer.

It is claimed that this valve design permits operation with a lower injection air pressure than with other types of atomizers. In practice it would appear that this advantage does not exist as at full load a pressure of about 900 lb. gage is necessary.

This design of spray valve is superior in many respects. Of special advantage is its ability to handle successfully, the heaviest of Mexican oils. Unlike many spray valves this one is able to atomize heavy oils without the formation of carbon on the piston crown.

Adjustments Valve Stem.—The valve stem, which is steel, corrodes with a sulphurized oil and must be burnished at each removal. The stuffing-box is best packed with shredded babbitt, although vulcanbestos may be used in an emergency. After the valve has been ground, it becomes necessary to bring the distance from the valve tip to the rocker lock washer back to the original length. This is accomplished by running both the washer and lock-nut up until there is the proper clearance between the fuel cam and rocker roller. The spring tension can be made normal by the insertion of a washer between the spring and the dashpot, or spring cap. The valve can be reground until the lock-nut is against the dashpot at which time a new valve must be secured.

Valve Cage.—The valve cage is held by two studs and rests on a taper seat in the cylinder-head casting. This taper joint should be cleaned of soot before the cage is replaced or the joint will leak.

In removing the valve cage the entire rocker assembly of all the valves must be lifted. When only the valve stem is removed, it is merely necessary to unbolt the upper part of the cage or housing which contains the spring. The valve stem will slip through the fingers of the rocker.

Fuel Line.—The fuel line, as on all engines, fills with dirt and must be flushed with kerosene. There is no by-pass or relief valve on the fuel line. If the oil line becomes air-bound, the union at the valve cage must be unscrewed to allow the fuel line to clear. To avoid a messy appearance after a line has been emptied, a shallow pan can be constructed to receive the oil.

Starting Fuels.—For starting purposes, it is customary for the manufacturer to furnish a two-compartment fuel tank, one compartment containing the fuel oil which is used, while kerosene is placed in the other part. This kerosene is supplied to the engine in starting since it will ignite at a lower temperature.

McIntosh & Seymour Marine Diesel Fuel Valve.—The fuel valve of the McIntosh & Seymour Marine Engine, while using a Hesselman atomizer, has a cage and actuating rocker quite different from that described above. The cage rests in a bushed opening in the cylinder head. The rocker arrangement is shown in Fig. 208. The rocker arm, at one end, is pinned to the vertical push-rod while the other end carries a hardened steel button in contact with a dog which, in turn, raises the valve by its adjusting put. The fuel cam, etc., have already been discussed in the preceding chapter.

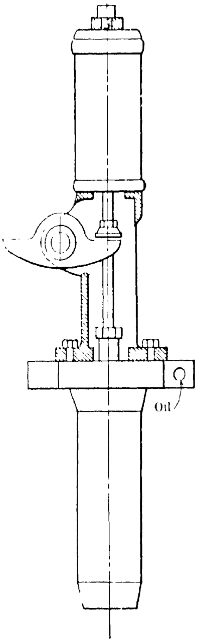


FIG. 208.—McIntosh & Seymour marine trunk piston Diesel fuel valve.

The Snow Diesel Fuel Valve.—The Snow horizontal Diesel fuel valve is of the open type and is shown in Fig. 209. The valve is enclosed in a cast-iron cage which is held to the cylinder head by two studs. The cage extension carries the valve spring, which seats on a cast-iron bushing about the valve stem. This cage has a gasket at the cylinder-head surface while a ground joint between the atomizer and head casting prevents water leakage into the cylinder. This arrangement provides an effective water-cooling of the valve cage without the necessity of a water cavity in the cage body.

The oil charge enters the valve cage at *c*, flowing along the check valve *D*, and comes to rest in the small cup or reservoir *E*. The air from the compressor enters at *B* and fills the recess at *A*. At the proper time the needle valve *H* is opened; the air rushes out the valve opening, through the passage *F* and into the cylinder at *I*. In sweeping over the surface of the fuel charge at *E*, the oil is picked up and, as it is forced along through the atomizer disks *C*, is broken up.

Valve Stem.—The valve stem can be removed, without disturbing the rocker arm, by unscrewing it from the spring bushing. In making adjustment for cam roller clearance, the stem is screwed in or out of this spring bushing as required. This allows a considerable shortening of the valve stem, from regrinding, before replacement is necessary. The lower part of the stem is grooved, and these grooves collect dirt and consequently should be cleaned occasionally.

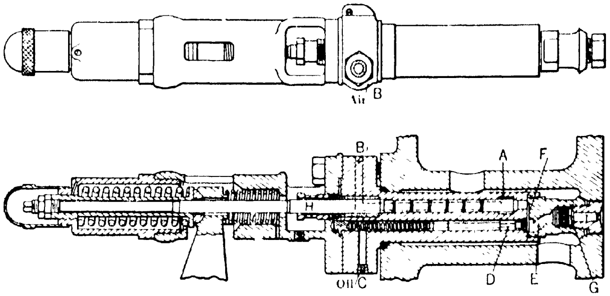


Fig. 209.—Snow horizontal Diesel fuel valve, open type.

Fuel Check Valve.—While cutting of the fuel valve seat due to dirty oil is avoided, the scoring is merely transferred to the check valve *D*. The oil is at all times in contact with this valve, and there is a strong tendency for the dirt to settle on the inner edges of the seat. This will cause the valve to leak, although the scoring action is much less than with the closed nozzle since there is no high-velocity air stream present.

Atomizer Disks.—The atomizer is provided with a series of fins about the peripheries of which are a number of notches, Fig. 210. These notches will fill with a tarry deposit when a heavy asphaltum base fuel is burned. The presence of these deposits is usually indicated by the engine requiring a high air injection pressure to maintain the correct speed. The atomizer disks should be cleaned at least once a month—more often if the

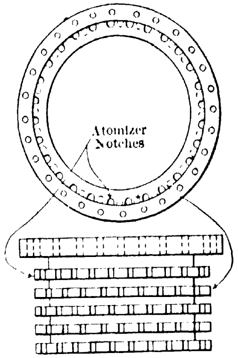


Fig. 210.—Snow atomizer disc.

oil is dirty. A spare set can be inserted and the old ones soaked in lye water.

Snow Vertical Diesel.—The vertical Diesel built at the Snow Works makes use of the mushroom type fuel valve employed on the Worthington Marine Diesel and shown in Fig. 211. The fuel valve-rocker arrangement is shown in Fig. 75. A rocker *D* is fulcrumed on the pin *E* which is supported by the end of the valve rocker *H*. The rocker *D* actuates both the fuel valve *F* and the air-starting valve *G*. The free end of this rocker fits into a groove *C* which is a part of the eccentric *B*. By moving the reach-rod *A*, the eccentric *B* carrying the groove *C* is shifted so that the groove *C* brings the rocker *D* in contact either with the fuel valve *F* or the air-starting valve *G* and thus enables the rocker *H* by moving *D* up and down to open either valve as desired.

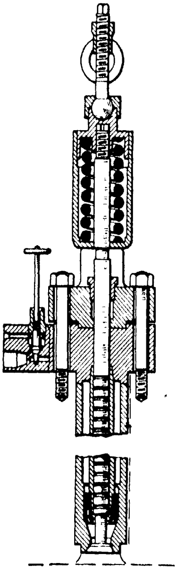


FIG. 211.—Fuel valve of Snow vertical engine.

The McEwen Diesel Fuel Valve.—Another design of open-type fuel valve is found on the McEwen Diesel. As outlined in Fig. 212, this valve consists of a cast-iron housing, or cage, which is bolted to the cylinder head with the axis of the fuel nozzle coincident with the axis of the cylinder, and of the necessary atomizer and needle valve.

The fuel oil enters the valve at *a* and, passing around the ball check, deposits in the fuel chamber *b*; part of the charge also flows into the fuel-valve plug. The air charge enters at *c* and surrounds the base of the valve stem at *d*. When the needle valve opens, the air passes through the valve opening and into the air port *e*; a part also completely fills the fuel-valve-nut cavity where it exerts a pressure on the oil contained in the reservoir *b*. The air flowing through the port *e* toward the cylinder attains a high velocity with a decrease in pressure. The air above the oil, having a fairly large volume, maintains its original pressure. This unbalanced pressure produces a flow of oil in the port *F*. The oil, after passing through the small channels *G*, encounters the stream of high-velocity air and is swept into the cylinder.

This oil issues from the channels *G* in very fine streams, which are quickly mixed with the air. The atomization is further increased by the action of the atomizating plate *H*. This valve is one of the best yet designed. The air does not "slug" the oil, causing a poor atomizing effect. On the contrary, the channels *G* are of such cross-section that the oil issues from them in streams of a size that will allow a thorough breaking-up by the air.

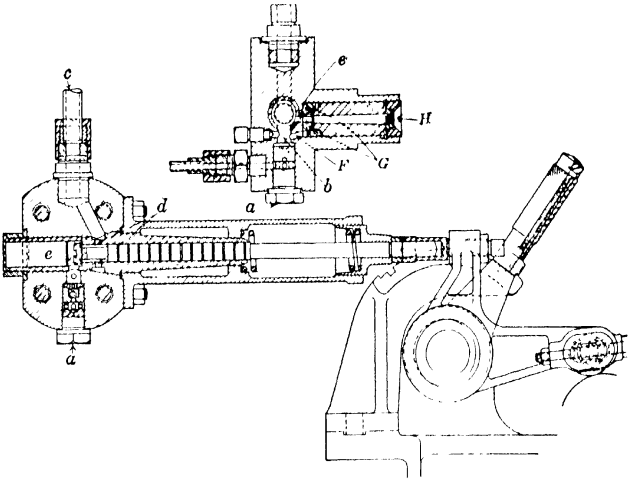


FIG. 212.—Fuel injection valve, McEwen Diesel.

Regardless of the engine load and the volume of the fuel charge, the rate of fuel injection into the cylinder is fairly constant with any given fuel. Even a decrease of 100 lb. in the injection air pressure apparently has but minor effect on the degree of atomization. Oils of different characteristics do not flow through the passages at identical rates. The natural consequence is a variable rate of fuel injection unless disks with different size passages are employed. A partial remedy lies in the control of the injection air pressure.

Adjustments.—The ball check valve requires some attention; especially is this true with dirty oils. In case the seat is worn, a new contact can be formed by using a hardwood stick or soft copper pin; lightly striking the pin will cause the ball to renew the curvature of the valve cage seat.

The roller, actuating the valve, is mounted on an eccentric. A lever moves the roller into and out of contact with the fuel cam. In changing the valve timing, the rocker roller can be moved by means of the adjusting pin. The adjusting pin passes through the roller pin or bearing and moves the bearing along the slots shown. The fuel valve rocker set-screw, resting on the end of the valve stem, controls the clearance between stem and rocker.

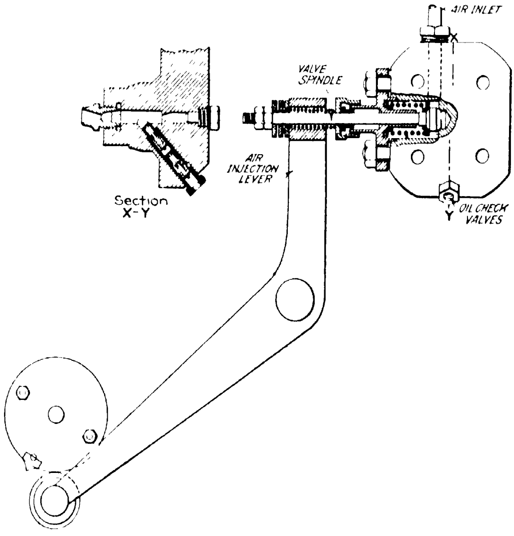


FIG. 213.—Allis-Chalmers fuel valve.

Allis-Chalmers Diesel Fuel Valve.—The fuel valve of this engine is of the open type, Fig. 213. The body is a steel block, the nozzle extension of which fits into the cylinder head. The oil enters the valve body at the bottom, the line having two poppet check valves. The air enters the block at the top and fills a recess behind the fuel valve. As the lever or rocker arm lifts the valve, the air rushes through the valve opening and flows into the oil cavity at the point *a*. As it passes over the body of fuel, the air picks up the oil and blows it into the cylinder. The velocity of the air stream, together with the action of the atomizer, breaks up the charge into particles. The nozzle tip is flaring in contour, causing the mixture of air and oil to assume a cone shape.

Adjustments. Check Valves.—The oil check valves require regrinding at intervals, especially with dirty oils. The check-valve cage is quite easily removed for valve repairs by unshipping the oil pipe line and unscrewing the cage from the fuel valve body.

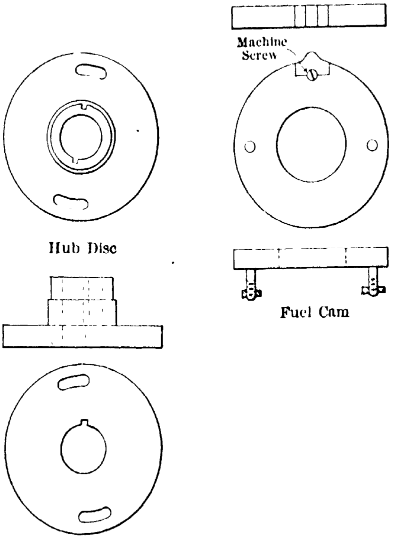


Fig. 214.—Allis-Chalmers fuel cam.

Needle Valve.—The valve is of generous dimensions, and consequently very little regrinding is necessary.

The Atomizer Tip.—The atomizer, since it is exposed to the cylinder temperature, carbonizes or gums from the small amount of oil that adheres to the atomizer after the air blast ceases. A spare tip should be on hand at all times. The presence of deposits will ordinarily be indicated by the engine laboring until the injection air pressure is raised.

Fuel Cam.—The layshaft is provided with a hub disk which is held by a nose key. This hub is fitted with an extension upon which is keyed the air-starting cam. The hub flange, as will be observed in Fig. 214, is slotted. Into these slots are placed the fuel cam bolts. The fuel cam also appears in Fig. 213, where it can be seen that it has an adjustable nose. In setting the fuel valve timing, the fuel cam can be moved several degrees around the hub disks and held in place by the bolts. This provides the easiest method of adjusting the fuel valve timing.

Timing the Valve.—In timing the valve instead of marking the positions on the flywheel a brass circular dial may be placed on the end of the layshaft, as shown in Fig. 215. A pointer is fastened to the rocker fulcrum pin. The dead-center positions of the piston is found by the method outlined in Chapter XII, and marks are made on the dial in line with the pointer. The dial may then be laid out in degrees over the arc covered by the firing period. Since the layshaft is run at a 1 to 2 ratio there is never any question as to whether the piston is on the firing or suction dead-center. If the engine is a twin-cylinder unit, two dead-centers and two arcs are laid out as shown in the illustration. These may be marked 1 and 2. If a three-cylinder, obviously three marks at 120 deg. are made. For a double unit the dials are placed on each layshaft.

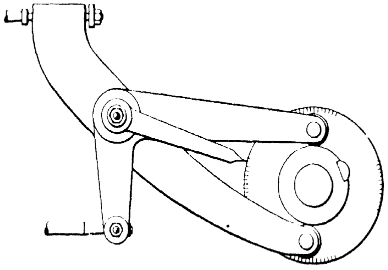


Fig. 215.—Timing Allis-Chalmers fuel valve.

The National Transit Diesel Fuel Valve.—The National Transit Co. equipped their Diesels with an open-nozzle fuel valve, Fig. 216. The valve stem is placed vertically in the valve block and has the closing spring located in the top of the valve body. This spring is enclosed in a sleeve *B* which acts as the air valve and is moved vertically by the valve stem compressing the spring, as the stem is raised by the rocker. The oil enters the valve passage of the atomizer through the check valve *a*. The oil reservoir extends the depth of the cylinder head. This causes the oil charge to spread out in a thin sheet; consequently, the air does not pick up the oil in slugs. Owing to this passage length, no atomizer disks are necessary.

The valve body has no cooling-water passages since it rests in a thin bushing pressed into the cylinder head; the cooling effect through the bushing is quite satisfactory.

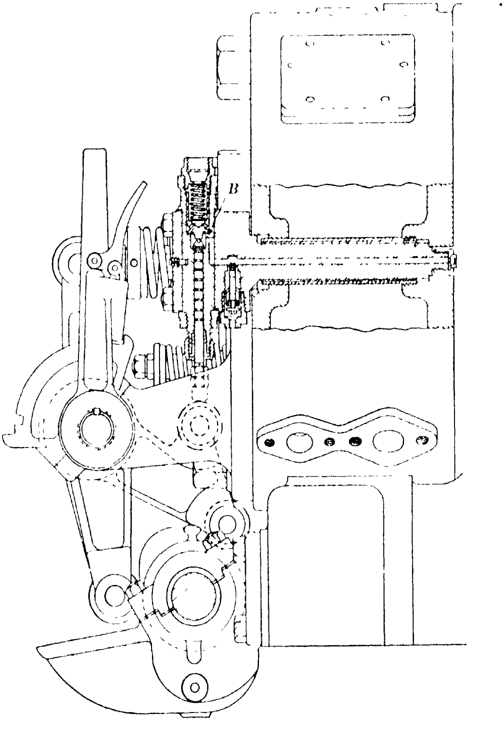


FIG. 216.—National Transit Diesel fuel valve assembly.

The fuel valve of the National Transit Diesel manufactured prior to 1918 was opened by a push-rod carried in the hollow housing of the camshaft end bearing. This push-rod is in contact with the fuel cam, Fig. 21.

On later Diesels the fuel valves are driven from the camshaft in front of the cylinder, as appears in Fig. 216. The fuel valve and starting-valve rocker are here mounted on an eccentric which is machined on the fulcrum shaft. In starting, a lever gives an angular displacement to the fulcrum shaft; this shifts the eccentric, throwing the fuel-valve rocker away from the valve stem and moving the air-starter rocker into position.

Fuel Valve Timing.—In timing the valve opening and closure the adjusting nut on the valve stem is set to contact with the fuel cam and the valve at the desired piston angle for valve opening, as with practically all engines. After the valve is reground a few times the center may touch the top of the valve stem, holding the valve open at all times. To overcome this trouble the center should be ground off at the surface where it seats on the valve stem.

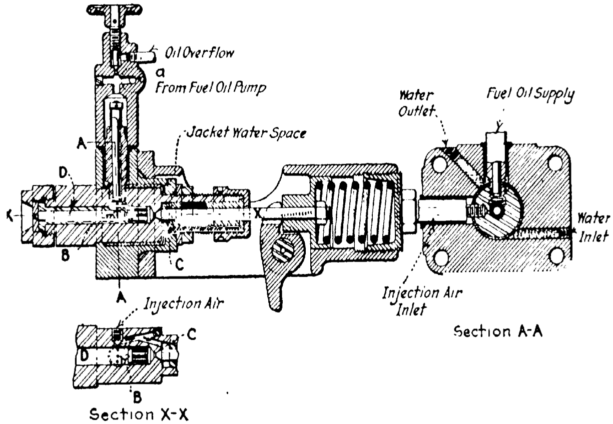


FIG. 217.—Standard horizontal Diesel fuel valve.

Standard Fuel Oil Engine Fuel Valve.—The fuel valve of the horizontal engine is of the open-nozzle type. The valve is housed in a cast-iron bracket block and consists, in the main, of the steel valve body, atomizer, needle valve, fuel check valve, and valve actuating spring and rocker, Fig. 217.

The operation is as follows: The charge of fuel, flowing through the pipe *a* and the check valve, enters the atomizer through the small port holes *B*. The air enters at the side of the valve block, passing around the valve stem above the seat at *C*. As the needle valve is raised, this air enters the atomizer passage *D*; to do this, the air must pass through the atomizer cone shown in the section *B–B*. This effectually breaks up the air stream, preventing the air from "slugging" the oil into the cylinder. As the high-velocity air passes over the oil, the latter is picked up and swept into the cylinder. As a means of giving sensitive control of the fuel supply on low loads and of eliminating "hunting" by the governor, a by-pass fuel valve is placed in the fuel line. When low loads are carried, the governor, which is of the Rites Interia type, is at the extreme limit of its travel. In this condition this type of governor is unstable and will hunt excessively. By cracking the overflow valve to allow part of the fuel pumped to by-pass back to the tank, the governor is compelled to handle more than the engine's requirements. The governor then moves outward to a more stable position, giving a closer regulation to the engine speed.

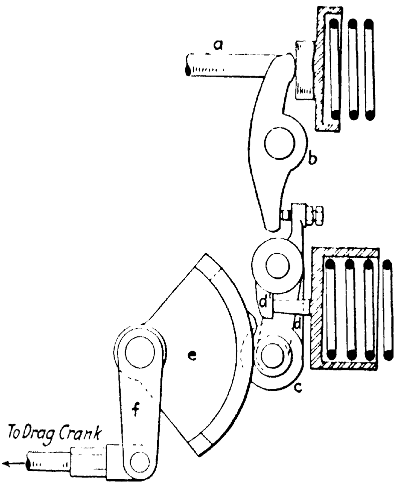
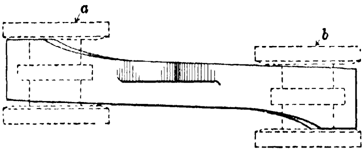


FIG. 218.—Standard valve gear

Fuel Valve Actuating Mechanism.—The fuel valve is operated by a unique cam arrangement radically different from that used on all other Diesels. The device appears in Fig. 218 where *a*, the fuel valve, is moved by the dog *b*; this dog, in turn, receives its motion from the cam lever *d*. On the lower end of this lever is mounted a roller which is in contact with the cam *e*. It will be observed that, as the engine turns over, the drag crank, being connected to the reach-rod and crank *f*, will raise the needle valve through this system of levers. The cam nose comes under the roller twice in each revolution of the engine, and, since the engine is a two-stroke-cycle, it then becomes necessary to provide some means whereby the roller is raised only once per revolution of the engine shaft. This is accomplished by equip-



Developed Surface of Cam
Showing Roller in the Two Extreme End Positions

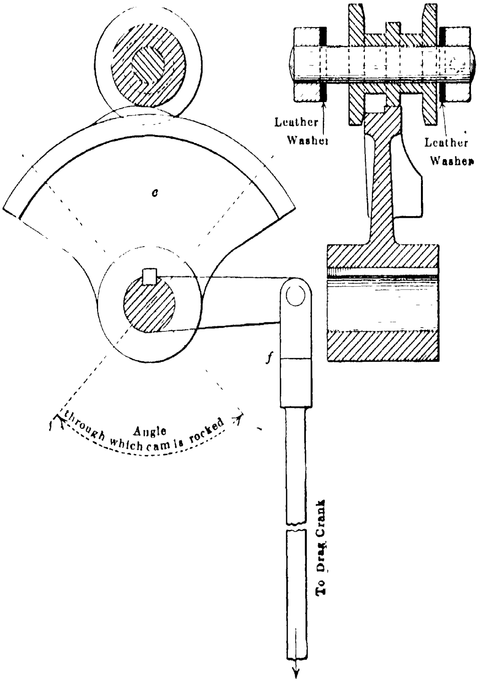


FIG. 219.—Standard Horizontal Diesel fuel cam.

ping the cam roller with two flanges, as more fully illustrated in Fig. 219. The cam is milled with curved sides, and the nose extends over only one-half of the cam width. As the reach-rod moves to the left, Fig. 219, the right-hand roller flange bears against the cam edge; the cam nose then passes between the flange and the roller. At the extreme travel of the reach-rod, the curved side of the cam throws the roller to the left, pulling it into the position *a*, Fig. 219. When the drag crank reverses the travel of the reach-rod *f*, the cam nose strikes the roller, raising the needle valve; the cam at the end of its travel then shifts the roller to the right to the position *b*, which allows the roller on the return stroke to slide over the cam, untouched by the nose.

Adjustments. Timing of Fuel Valve.—The point of beginning of fuel injection can be altered by adjusting the reach-rod length. The clearance between the valve spring cap and dog is controlled by the adjusting set-screw on the roller lever *d*; this need not be more than $\frac{1}{32}$-in.

Fuel Valve.—The valve body is water-cooled, and it is necessary to use this cooling system when high-gravity fuel oil is burned. With heavy oils, around 24 deg. Baumé, the water tends to chill the valve and lower its atomizing efficiency; the result is a decidedly smoky exhaust. It is advisable, with such oils, to discontinue the flow of water around the valve.

Needle Valve.—The valve seat requires the same attention as do all open-type valves. The shortening of the valve, due to regrinding, can be compensated for by screwing the valve out of the spring cap or dash-pot.

Fuel Check Valve.—Especial attention should be given to the fuel check valve while the plant is burning dirty oil. If the fuel be sulphurized to any extent, this valve will corrode and of course demand regrinding. The check valve stem must be kept clean from corrosion since the spring is light and will not close the valve against any decided binding of the stem.

Standard Vertical Diesel Fuel Valve.—The vertical Diesel built by the Hadfield Penfield Co. makes use of a closed-type fuel valve. As indicated in Fig. 220, the valve assembly consists of a cast-iron housing into which is fitted a bushing having two grooves, one for the oil and the second and top one for the air passage. Inside this bushing is slipped a second bushing or sleeve through which the needle valve moves and whose upper end is provided with a stuffing-box packed with babbitt shavings.

The flame plate is provided with a number of drilled passages which converge at the lower end into a central nozzle opening. This plate is held in place by a screwed cap.

The dashpot and spring are carried in a housing bolted to the fuel-valve body. The needle valve is actuated by a rocker arm fulcrumed on the upper housing. The camshaft runs along the side of the cylinder head and is driven by a vertical shaft meshed with a helical gear on the crankshaft. The needle value of a 50-hp. cylinder is ½ in. in diameter, and, although it is claimed by many that so small a cylinder is impossible in a Diesel, the combustion in this engine is as good as in larger Diesels.

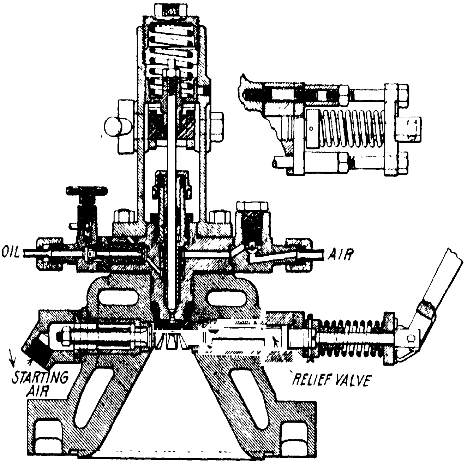


FIG. 220.—Standard Vertical Diesel fuel valve.

The spraying of the fuel is all that can be desired over a wide range of speed. The atomizing at 200 to 350 r.p.m. is practically uniform.

The oil line is provided with a drain cock so that priming is assured. The oil-line check valve is immediately at the fuel valve housing, thereby eliminating all danger of back fires. Being provided with a slotted head and carried in a housing or cage, the check may be readily reground.

Fulton Machine Co. Diesel Fuel Valve.—Figure 221 illustrates the fuel valve used on the three Fulton Marine Diesels built several years ago. It is along standard practice in closed nozzle valves and is controlled by a rocker from the camshaft. The engine is no longer manufactured, but the spray valve is interesting as showing that even in the smallest engines (50 hp. per cylinder) the general design of valve successful in large engines is used with no difficulty.

New London Diesel Fuel Valve.—The fuel valve used on the New London Diesel is shown in Fig. 222. The chief thing distinguishing this valve is the absence of atomizing disks, only a sleeve being used. In addition, the flame-plate carries only one opening instead of a number as is customary. Due to the compact form of the combustion chamber, as will be noticed in Fig. 190, it is not necessary to have the oil spray diffused. The needle valve is actuated by a rocker, as will be noted in Fig. 190, which is connected by a reach-rod to a cam rocker. The fuel cam rocker is mounted on an eccentric pin, which allows the valve to be thrown out of action.

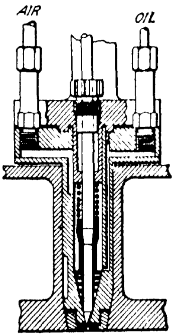


FIG. 221.—Fulton Machine Co. Diesel fuel valve.

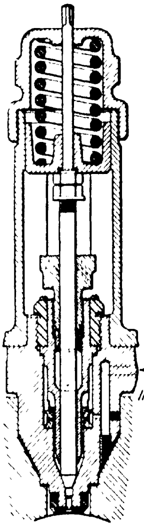


FIG 222.—New London fuel valve.

Nordberg Diesel Fuel Valve.—The spray valve Fig. 223 used on the large Nordberg Diesel is also found on the smaller type VE engines without change in design save that the former has a small rocker that serves to transfer the motion of the large rocker to the needle valve. This rocker is necessary in the large engines since the large rocker has its roller resting on top of the fuel cam rather than under or between the camshaft and cylinder head. On the small units the camshaft is below the cylinders, and the rocker is actuated by vertical push rods, the rocker bearing directly upon the needle-valve lock nuts.

The fuel valve body fits into an opening in the cylinder head, the water being sealed off by a thin bushing. Within the valve housing is placed a long pulverizer tube which carries a set of perforated disks at its lower end. The fuel flows down the clearance space between the body and the tube while the air flows between the needle valve and tube, emerging through drilled passages above the disks. The flame plate carries one central nozzle opening with flaring exit walls, thereby enabling the hollow cone of oil, formed by the flow along the conical needle valve end, to spread over the combustion space.

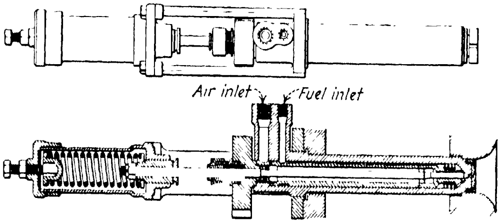


FIG 223.—Nordberg fuel valve.

Fulton Iron Works Fuel Valve.—This fuel valve, Fig. 224, does not differ in the atomizer design from the majority of fuel valves. The needle valve at its upper end screws into the spring holder and is locked into place by the jamming of an adjusting screw bearing on the end of the needle. Spring tension is adjustable over a wide range.

The needle is lifted by a short rocker fulcrumed to the upper cage casting. A roller is interposed between this rocker and the large rocker arm. The roller is linked to a shaft passing along one side of the set of fuel valves. On light loads the operator by moving a handwheel on the shaft causes the roller to move out along the short rocker. This results in the movement of the large rocker arms giving a smaller lift to the needle valve. As a consequence, the opening around the needle valve may be varied with the load; in addition the shape of the roller track on the short rocker is such that the clearance between roller and large rocker is greater on low loads. This feature allows the duration of the valve lift to be varied, covering a smaller crank angle on low loads. The clearance with the roller at the inner point of its travel is about 0.03 in.

It has been found that on light loads the amount of oil entering the spray valve is so small that all the top atomizer disk surface is not covered, the oil piling up on the side next to the oil passage. There is then a loss of air blowing down the opposite side. To eliminate this the oil passage was changed so that it now flows down a passage cut into the outer surface of the sleeve *O*. The lower end of this groove is divided into two passages, allowing the oil to enter the cavity at two opposite points. This gives a better distribution of the oil.

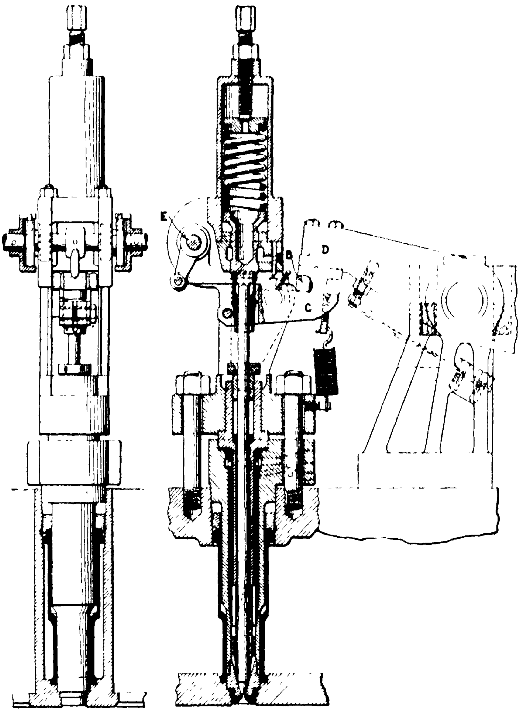


Fig. 224.—Fulton fuel valve assembly.

Lombard Fuel Valve.—This fuel valve is outlined in Fig. 71*A*. It is somewhat unusual in that, while set vertically in the cylinder head, the nozzle has a side opening. The fuel is introduced into the cylinder in a fan-shape stream horizontally across the piston crown. The stream of oil particles in this way comes in contact with the air in the combustion space and by reason of its thin fan shape does not strike the piston crown or cylinder head. It is claimed that this encourages perfect combustion since none of the oil is chilled by the cold iron walls. In operation, however, it is evident that a small amount of the oil does touch the walls since the exhaust is not clear until the engine warms up. It would seem that this character of oil stream should be superior to the usual cone.

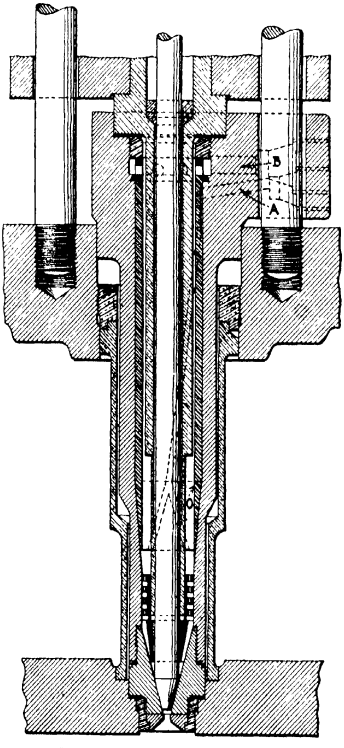


FIG. 225.—Fulton fuel valve.

Worthington Marine Diesel Fuel Valve.—The large marine Diesel built by the Worthington Pump & Machinery Corporation makes use of a mushroom-type fuel valve in preference to the usual needle valve, as shown in Fig. 225*A*. The mushroom valve has been proved to be successful by one of the largest European builders. It has the advantage of insuring a thorough distribution of the oil stream since in its passage between the inwardly opening valve and the valve seat the oil assumes a cone formation. The disadvantage of this type of valve for small cylinders is that the height of the opening between the valve and seat must be small since the length of the opening is necessarily large. The difficulty of adjusting the valve lift to reduce the air loss is evident. On large units these objections do not appear. The

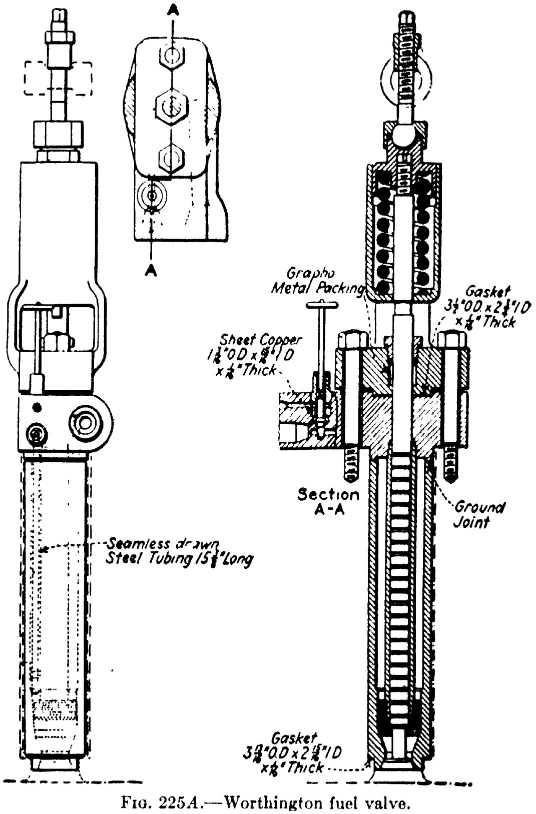

A
A
Grapho Metal Packing
Gasket 3½"OD x 2⅞"ID x 1/16" Thick
Sheet Copper 1¼"OD x ⅝"ID x 1/16" Thick
Section A-A
Ground Joint
Seamless drawn Steel Tubing 15⅝" Long
Gasket 3 3/16"OD x 2 15/16"ID x 1/16" Thick

FIG. 225*A*.—Worthington fuel valve.

air enters the valve cage between the pulverizer tube and cage while the oil flows down a copper tube in a drilled passage in the cage wall and empties immediately above the atomizer disks. The rocker arm tappet bears directly on top of the spring cap, the rocker being actuated by a push rod from the cam shaft which runs along at a level with the cylinder bases.

Cramp's B. & W. Diesel Fuel Valve.—This fuel valve is shown in Fig. 226. The valve has a mushroom head and the construction is in general similar to the one just discussed. In regrinding this type of valve the cage is removed, and the process is quite like that followed on exhaust valves. The contact need be but a hair line.

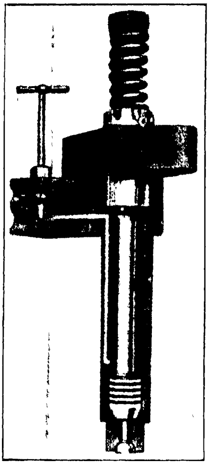


FIG. 226.—Cramp's B. & W. fuel valve.

Twin Fuel Valves.—In Fig. 227 is shown the arrangement of twin fuel valves on the 1,000-hp. U. S. Navy submarine Diesel. This engine was built along the lines of the German U. Boat Augsburg Diesel where a m.e.p. of 150 lb. per square inch was attained. Many designers and engineers hold the belief that twin fuel valves were first used on the Augsburg to secure better fuel distribution and combustion. In fact, the underlying motive was to obtain a better distribution of the heat stresses in the cylinder head. It has not been proven by any experiments that two fuel valves give better combustion.

The two valves are lifted by the spider *B* which is controlled by a forked rocker arm hooked to a pin passing through the bore *I*. The spider is guided vertically by the holders *A* and *H*. The spider bears on the bushing *D* which in turn is held down by the large spring *C* which also holds the small spring *E* in compression as soon as the motion of the spider compresses the large spring *C*, the spring *E* is relieved and raises the needle valve. The possible lift of the needle valve is regulated by the nut *F* and sleeve *G*.

Maintenance of Fuel Valves.—Superior advantages are claimed for each type, but from the point of view of maintenance there is little choice. The spray-valve troubles most commonly encountered are scoring and cutting of valve and seat, causing valve to leak; jamming of stem in its guide, preventing valve from seating; valve-stem packing leaking; and atomizer plates or passages clogged, preventing flow of oil and air out of valve body. In the case of a leaky seat or valve its correction may involve more or less regrinding, depending on the depth of the cuts or scores on valve and seat. In no case should recutting or facing be restored to if it is possible to make the valve tight by grinding, but it should be remembered that frequent regrinding is likely to result in the seat wearing out of true or the valve face becoming uneven.

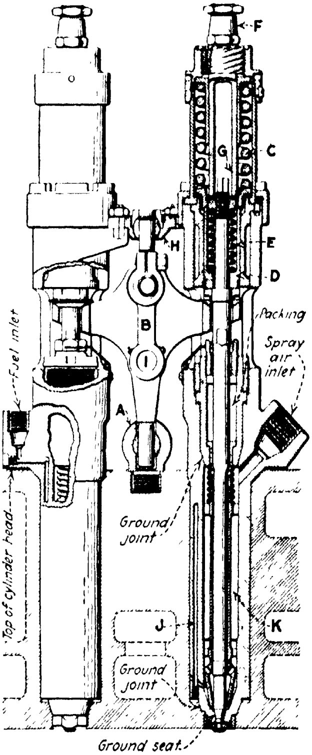


Fig. 227.—Twin fuel valves of 1,000 hp. U. S. submarine Diesel.

Regrinding Fuel Valves.—Regardless of the type of fuel valve, each, sooner or later, requires regrinding. In performing this process, an engineer should be very miserly with the amount of grinding paste used. The best compound is one of powdered glass and vaseline, or emery flour and vaseline. Only a very small amount should be placed on the needle, being spread out evenly over the entire seating surface. The entire valve, with the exception of the spring, should be assembled when grinding. This is to insure that the valve is aligned properly. It is unnecessary to secure more than a thin line contact at the seat—$\frac{1}{64}$ in. in width is ample. After grinding, it is advisable to disassemble the entire valve and cage and wash very thoroughly with kerosene. This is to remove all emery particles.

It is impossible to lay down a hard and fast rule as to the frequency of regrinding. If the fuel is fairly clean once every 800 to 1,000 hrs. of service seems to be about correct. Poor filtering equipment may cause valve leakage much before this.

Some engine builders supply a special reamer for trueing the spray-valve seat, but if it is not supplied, the engineer should have one made. The reamer should be a duplicate of the valve, with the face fluted to form cutting teeth. The reaming should be very carefully done to obtain good results. The reamer should not be dropped down on the seat nor should very much pressure be applied, as it will chatter and cut an irregular seat. Insert the reamer carefully, hold against the seat with a light pressure and turn only in one direction. If it is turned backward the chips collected between the cutting teeth will scratch the valve seat and a great deal of grinding will be required to remove the scratches. The liability to scratch will be reduced if the cutting teeth are coated with heavy cup grease. This grease will catch and hold the metal chips and prevent them being dragged under the teeth. Remove the reamer frequently, clean and recoat with grease, wipe out the valve seat and continue the operation only long enough to obtain a good full seat, without removing any more metal than is necessary.

It is general practice to make the valve seats of cast iron and the valves of high-grade alloy steel, hardened. For this reason it is usually the valve seat that requires recutting and the valve need be ground only to the seat, but when the valve is scored enough to require recutting it is best done in the lathe, using a center grinder in the compound rest as a cutting tool. The rest is set to the angle of the valve seat and very light cuts are taken, using a soft grinding wheel and revolving the valve about 200 r.p.m.

New Needle Valves.—The author, at one time, in view of the unsatisfactory needles obtainable, recommended that the engineer make all needles required for replacement. Because of the improvement in needles supplied by the builders it is now conceded that only in case of emergency should an engineer attempt to do this. In all save exceptional cases the engine builders' parts prove more satisfactory. The valve is capable of considerable shortening by grinding before it is useless. The greatest cause of replacement is the corrosion of the stem due to acid in the oil. In addition, there is some wear at the stuffing-box gland by reason of the pressure of the packing.

Needle Valve Wear.—The spray-valve stem guide is usually made as long as possible in order to reduce the wear and insure the valve seating absolutely true. It is a common mistake among engineers to assume that sticking of the valve stem in the guide is due to a too close fit between stem and guide. As a matter of fact it is in most cases caused by this fit being too loose. The spray air pressure inside the spray-valve body tends to force the fuel oil up into the space between the guide and stem, especially when leakage of air past the valve-stem packing creates a condition of unbalanced pressure in the valve-stem guide. This fuel oil is spread in a thin film over the surface of the valve stem, and the heat of the stem gradually cooks it down to a gummy condition. This gum or soft carbon is very gritty, with no lubricating value, and will soon cause the stem to stick in its guide. When the spray valve is disassembled for overhaul, the stem and guide should be carefully cleaned with kerosene. Emery cloth should not be used to polish the stem unless it is too rough to be cleaned with oil, as it tends to reduce the diameter of the stem and increase its clearance in the guide.

Spray-valve stems are packed with high-grade soft packing or metallic packing. In replacing, do not use packing that contains rubber. Rubber is readily solvent in fuel oil and the packing will be destroyed rapidly. If the stem shows cuts or abrasions where the packing bears, it should be smoothed and polished with emery cloth to prevent the packing from being torn.

Bent Needles.—If the valve stem is bent, it should be straightened and spun between centers to see that it runs true. Do not replace a stem that has been straightened without testing the seating of the valve. If the spring has taken a permanent set that shortens it enough to decrease its working tension place enough washers under the spring or on top of it to give it the right tension. Since it is not practicable in the ordinary engine room to measure this tension, the engineer must necessarily set it by observing the action of the valve. The tension should not be so great as to put unnecessary load on the operating gear, but it should be enough to overcome the inertia of the valve levers and push rods and permit the valve to snap rapidly back to its seat after the pressure of the valve lever is removed.

Lubrication of Valve Stem.—The valve stem gradually becomes coated with a thin layer of oil residue, this being more noticeable in non-cooled valves. To prevent binding of the stem, it should be constantly lubricated. A small amount of kerosene injected around the valve at least once every 24 hrs. will remove any residue.

Leaky Fuel Valves.—A leaky injection valve usually betrays its presence by causing the engine exhaust to be smoky. Leaky valves also allow the fuel charge to seep into the cylinder during the compression stroke and so produce violent preignition.

Incorrect Fuel Valve Timing.—If the valve opens too early, a sharp metallic click or pound will be heard in the cylinder. This is evidence of premature combustion. If the valve opens late, a dull thump or pound, quite like a pound due to loose pin bearing, can be heard. Furthermore, a smoky exhaust ordinarily accompanies this pounding.

Clogged Atomizer or Nozzle-tip Disk.—When either the atomizer or the disk at the end of the fuel valve, sometimes called the burner plate, is clogged, the exhaust is smoky.

Sticking Valve Stem.—When the fuel valve stem sticks in the open position, the exhaust will be smoky and the injection air gage will show a drop—with the open nozzle, the gage needle will show as much as a 75 per cent pressure drop.

This sticking may be due to a gummed stem or to excessive packing friction. The engineer is cautioned against allowing such a state of affairs to exist. There is great danger that the flame in the cylinder will flash back through the fuel nozzle since the air pressure will be low by reason of the free passage into the cylinder. This "flash back" will burn the flame plate or nozzle and the needle tip. In extreme cases the explosion has wrecked the full valve cage and air lines. If the edges of the flame plate orifices seem to be burned, as in Fig. 228, it is proof that there has been a "flash back."

Buried Needle Tips.—If the needle spring is too heavy or too light in compression, in closing the needle will chatter. Ultimately this will lead to a badly worn seat in the valve body. To rectify this operators will deream the seat. If rereaming is continued, there is a likelihood of the needle seating so deep in the body, as shown in Fig. 228*A*, that when it is lifted the round part is still below the top of the seat cone. If this happens the fuel cannot pass by the needle stem and down the seat. The result is that the entire oil line to this spray-valve casting is filled with oil. On the next pump stroke the pump plunger comes up against this solid mass of oil and something must give way. If the governor acts directly on the plunger, the governor becomes irregular. In other cases the oil line must relieve itself of leaking at the joint or the high pressure may raise the needle valve at the wrong time. The danger in con-

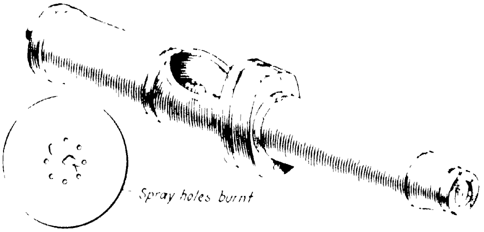


Fig. 228.—Burnt flame plate.

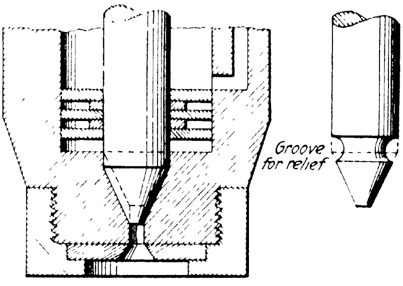


Fig. 228*A*.—Buried tip.

tinued rereaming is obvious. As a remedy it is recommended that a groove be cut around the stem at the juncture with the tip. This groove will provide a clearance for the oil when the needle lifts.

Fuel Valve Cooling Jacket Temperature.—The desirable temperature, at which the discharge line from the fuel valve water jacket should be carried, depends on the characteristics of the fuels. If the oil is heavy and viscous, the discharge should be around 160 deg. F. With fuel oil of 28 deg. Bé., and higher, 120 deg. F. is amply high since the valve must be cool to prevent gassing of the light oil.

Adjustable Injection Air Pressure.—The engineer can appreciate the necessity of having a higher injection air pressure when the engine is carrying full load than when under a light load. When the fuel charge delivered to the fuel valve is large, as on full load, the resistance or "braking" action of the atomizer is high and requires a high pressure to force the entire charge of oil into the cylinder. On light loads, the oil occupies only part of the atomizing space, and consequently a light air pressure is sufficient. If the pressure is high on light loads, the oil is blown into the cylinder at an increased rate. The passage of the fuel would then require only part of the time during which the needle valve is opened. The remainder of the period of valve opening would be devoted to the passage of pure air. The high velocity of the free air as it left the nozzle tip would chill the tip and lower the entire cylinder temperature, causing a decreased cylinder efficiency as well as a direct loss of air that has been compressed at a considerable expense of power. Furthermore, if the air pressure is high on low loads, a sharp knock is produced in the cylinder which results from the inrush of air at a pressure far above cylinder pressure. Conversely, if the air pressure is too low the engine will smoke since the fuel has not been sufficiently atomized.

It is necessary for the successful operation of any Diesel that the injection air pressure be altered to conform to load change. This adjustment can be under manual control of the engineer, as is the general practice. The manual control can be obtained in several ways. The McIntosh & Seymour Marine engine and the crosshead type stationary engine are provided with a clearance chamber on the low-pressure cylinder, whose volume can be altered, changing the air discharge pressure. Other builders arrange for the operator to adjust the low-pressure suction, obtaining the required air-pressure control. However, on fluctuating loads, this entails constant attention and is more suitably handled by some automatic arrangement. There are several designs of automatic injection control. The Busch-Sulzer Diesel throttles the compressor suction through a linkage from the engine governor. The Standard Fuel Oil engines, as has been outlined, use a governor-controlled air by-pass valve.

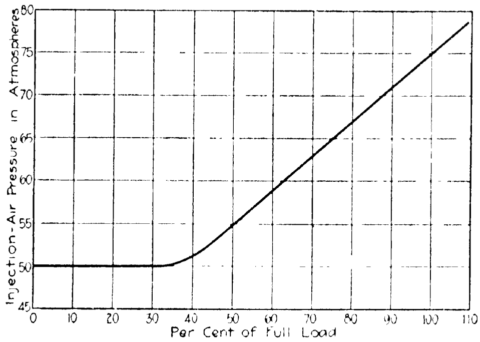


FIG. 229 Injection-air pressure.

Injection-air Pressure.—Each design of fuel atomizer requires a certain air pressure to inject the oil properly. This pressure will vary with the resistance or "braking" action of the atomizer disks. It follows that no set pressure values can be given which will apply to all engines. However, the curve in Fig. 229 gives the relation between engine load and injection pressure that is correct for practically all closed-nozzle engines. For open-nozzle units the pressure will run some 50 to 100 lb. lower than is given in the table. In making this statement it has been assumed that the difference between injection-air pressure and cylinder pressure was a constant. This holds perfectly true for similar types of engines, but is not true for such widely different types as a high-speed submarine engine and a low-speed heavy-duty stationary or merchant-marine engine. Both types use about the same cylinder compression pressure (about 475 lb. per square inch), but the combustion pressure in the high-speed engine rises to perhaps 550 or even 600 lb. per square inch, while in the heavy-duty engines it is only 550 lb. or even no more than the compression pressure. The best injection-air pressures are found to be from 1,000 to even 1,200 lb. per square inch for the high-speed engines and generally about 800 lb. per square inch for the low-speed engines. These differences are due to the conditions of working. From the high-speed engine the highest possible power output for a given piston displacement is desired, while in the case of the low-speed engine, reliability and economy are the main objects sought.

In any case the lowest possible injection-air pressure that will give the results wanted is the best. The lower the pressure, the easier the duty on the air compressor and the less the chilling effect on the combustion in the cylinder, due to the expansion of the air. With a new type of engine the best method to be pursued is to run a series of tests with gradually increasing sizes of nozzles and varying injection pressures with each nozzle. A point will soon be found where increased size brings no advantage, and that size with the proper injection pressure is the one to be adopted.

At full power the engine will run with a wide variation in injection-air pressure and the economy will not be badly upset unless an extreme pressure is used; but for light loads the available range of pressure is very short because too great a pressure will give such a chilling effect that the temperature in the immediate vicinity of the fuel blast will be lowered to the point where no ignition will result, and too low a pressure will atomize the fuel. In fact, with a very low pressure there is danger of burning the spray-valve seat or even doing more serious damage. It is, therefore, never advisable to drop the injection-air pressure below 550 to 660 lb. per square inch under any circumstances. The available range of pressure at the low end of the scale is perhaps 50 to 100 lb. per square inch. For full-load conditions, if the very best pressure is, say, 850 lb., acceptable pressures would probably range from 825 to 900 lb. The economy is more seriously affected by a reduction in pressure than by an increase.

In order to overcome some of the difficulties just mentioned, certain engine builders arrange to have one-half of the cylinders cut out at very light loads. This increases the load on the other cylinders enough to give regular ignitions and not only is the engine less sensitive to a variation in spray-air pressure, but incidentally greater accuracy of governing is secured. Some builders even go a step farther and vary the spray-valve lift with the load, which practically eliminates any sensitiveness due to injection-air pressure variations. Under these conditions there is little difference between the best pressure at full load and at light load, but this is under the control of the governor and does not need to worry the operator.

Marine Diesel Air Pressures.—The marine engine case presents greater difficulties than the stationary since the propulsion power decreases approximately as the cube of the ship's speed and the revolutions only directly. It is customary to fit rather large air compressors in order to have a surplus capacity for refilling the starting-air tanks and also to provide a margin for reduced-speed running. As the speed is reduced, the total time that the spray valves are open during a second or a minute is the same, and consequently with the same air pressure the air consumption is constant in spite of the reduction in speed. The reduced air pressure which is used reduces the consumption to a certain extent, but since the production is only directly proportional to the speed of revolution, a point is reached at from one-third to one-half speed where the supply fails to keep up with the demand. By this time the power required has also dropped to such an extent that ignition troubles begin, so that this becomes the lower operating limit for practical purposes with constant time of opening injection valves. With special care and a skilled operator this limit can be lowered, but under ordinary conditions one-third speed is about the limit. In most cases this is no disadvantage whatever since the full speed is generally ten to twelve knots and one-third of these speeds is barely steerageway. To express it in another way, a marine engine operates at nearly full power and speed most of the time, and provided it can be slowed down on occasion and can be maneuvered satisfactorily, all requirements will be met.

Finding the Best Pressure.—The simplest method of determining the best injection-air pressure is to watch the exhaust temperature. Without a doubt, the exhaust-temperature thermometer is one of the most important instruments used around an oil engine, because no matter what happens to upset the combustion, the result is immediately registered on the thermometer. The operator may need all sorts of instruments and special tests to locate the trouble, but the exhaust temperature, if watched and compared, never fails to give a warning when something is wrong with the combustion. Consequently, the way to find the best injection-air pressure for a given set of conditions is to vary the pressure and watch the thermometer. When it registers the lowest, record the pressure for future reference and use; once determined, it will always hold until some condition is changed. The whole procedure is so simple that there is no excuse for an operator not knowing the best pressure for any condition. It would be sufficiently important if it were only a question of economy, but it means reliability as well. Poor combustion and the resultant high exhaust temperature invite trouble all around.

Adjustable Fuel Valve Timing—The usual Diesel engine fuel valve is designed with a constant period of valve opening, regardless of load conditions. In the Otto-type explosive engine the efficiency of the engine depends on the maximum explosive pressure. With the Diesel engine the efficiency depends both on the combustion pressure, which should be identical with the maximum compression pressure, and on the duration of the fuel injection. It is very clear that with load changes the time during which the fuel is injected should also vary. Since the rate of combustion should be constant, the period of injection must vary if the greatest possible efficiency is to be secured. Furthermore, a factor of operation also enters into the problem. On low loads the amount of oil is small and will be entirely blown into the cylinder long before the valve closes. The balance of the valve opening period is taken up with the injection of high-pressure injection air. This air assists in no way toward the combustion. For these reasons several builders, including the Busch-Sulzer Co. have adopted a form of injection-timing control.

Timing of Fuel Valves.—In timing a fuel valve the engine is pinched over until it is several degrees ahead of the desired point of fuel-valve opening. The air-line valve is "cracked," giving about 75 lb. air pressure on the fuel valve. The indicator plug is removed, and the engine is slowly barred over until the trammel cuts the opening mark on the flywheel. The injection valve should now start to open, as evidenced by the sound of injection air blowing into the cylinder. If the valve opens before the mark is reached, the rocker clearance can be increased, producing a later opening. If the valve opens late, the clearance can be reduced. The engine should be barred on to the closing mark, and the sound of the escaping air should cease as the mark is reached. Since the roller clearance has been altered to make the opening earlier, the closing point will probably be late. It then becomes necessary to turn the engine back ahead of the valve opening mark and shift the cam nose. The nose should be shifted to produce the required opening with the roller clearance correct. Then, on checking the closing point, it should either be correct or early. If the latter, the nose must be shifted back a trifle and the roller clearance made less. This should produce the required opening and closure. If the nose is excessively worn, it is impossible to obtain a correct timing, and a new nose must be secured. This method works very well with small engines, but in the case of a large engine in which the cylinder volume near the time of fuel admission is quite large relative to the area of the hole through the indicator cock, a slight movement of the piston causes quite a rush of compressed air through the indicator cock and it is not always possible to determine the exact point of spray air admission.

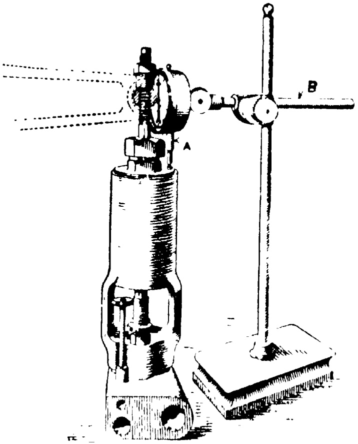


FIG. 230.—Setting fuel valve by dial gage

There are several other ways of determining the exact time the valve lifts from its seat. An approximately correct method is to start revolving the cam roller when the cam toe is just about to make contact. When contact between the cam and its roller has progressed sufficiently to jam the roller so that it cannot be turned, the valve may be considered open. This method is not accurate, as the clearance in pin bearings, deflection in levers, etc., may result in the valve leaving its seat an appreciable time after contact between cam and roller is made.

The most satisfactory method of all is to make use of a dial test indicator such as is generally employed in machine shops. This instrument is shown in Fig. 230. The instrument may be used on its base, or the arm *B* may be removed from its holder and clamped to any stationary part of the engine structure near the valve being set. Pointer *A* is placed in contact with the valve stem and any movement of the stem will be registered by the hand on the dial. The dial can be read to one-half of one-thousandth of an inch, so the exact point of opening of the valve is easily determined. By this method all error due to lost motion in the valve gear is avoided, as only the actual valve movement affects the indicator.

Having set the valves to open at the correct point as called for by the builder's valve diagram, the time of closing and total lift should be measured just as a matter of record. If the dial test gage is used, the maximum lift may be read directly from the dial. The same measurements should be taken for the astern cams.

Checking the Valve Settings with Indicator Diagrams.—After the fuel valves have been timed the engineer should apply an indicator to each engine. Any difference in the diagrams should be investigated for it is possible that variation in the cam outline or in the roller clearance is the cause of the difference. It is not an unusual thing for the spray valves on an engine to vary as to time of opening. If each valve is set to open, say at 5 deg. before top center, it may be found upon taking indicator diagrams that one or more valves may be slightly late or slightly early and it may be necessary to vary the times by as much as one degree in order to get a good combustion line.

Worn Cams.—Due to cam outline sometimes the rocker roller does not follow the cam but jumps, causing ridges beyond the cam nose. This is also traceable at times to the valve spring being too weak or to the stem sticking.

Worn Needle Seat.—If the valve spring is too stiff the needle will close with too great a blow followed by rebounding. This results in the valve seat pitting and wearing beyond hope of rereaming. Likewise with a weak spring there is often a tendency for the needle to delay closing and finally closing with severe blow. The engineer must experiment and determine the spring pressure that will cause the needle lock-nut to remain in contact with the rocker arm as the needle closes.

In case of wear of the seat the nose of the valve housing, if not of the renewable type can be cut off and a new nose made.

Back Lash.—The camshaft gears are not immune to wear, and in the course of 5 to 7 years of constant service the back lash between the gears becomes noticeable. The clearance between the gear teeth has a very detrimental effect on the injection cam. As the cam nose contacts with the valve rocker roller, the pressure that the spring offers against the rocker movement is considerable. As the roller travels over the surface of the cam nose and starts down along the back slope, this spring pressure forces the camshaft forward, causing the valve to close early. Since the wear is between the teeth, the camshaft is already behind its exact timing with the engine shaft, and consequently the opening of the fuel valve is late and the closure is early. A new cam nose of greater length will partially overcome the defect, but new gears should be ordered to replace the worn ones.

CHAPTER XIV

FUEL PUMPS

TYPES. ADJUSTMENTS

Fuel Pumps.—While adjustments of a Diesel fuel pump are not of frequent occurrence, nevertheless, this particular part of the engine is of vital importance. The successful operation of a Diesel depends, in a great measure, upon the accuracy and reliability of the pumping mechanism. When it is considered that on a 100-hp. cylinder, operating at 200 r.p.m. or 100 power strokes per minute, the volume of a single full-load fuel charge is less than 0.2 cu. in., the necessity of accurate pumping is apparent. Since the usual speed regulation requirement is 2 per cent on each side of normal, the extreme variation that is permissible in the volume of a single injection at a given speed is 0.004 cu. in. The pump, then, must be not only correct in design but also absolutely high-grade in the workmanship involved in its actual manufacture.

As has been previously outlined, two types of fuel injection are in use: the open- and the enclosed-nozzle type. The fuel pumps follow the same classification. First, the pumps on engines employing the closed-nozzle fuel valve must be constructed to resist a pumping head equivalent of more than 1,000 lb. per square inch. This demands rugged construction and absolutely leak-proof pump valves. The governor control, where the control is through the pump plunger, is called upon to withstand severe stresses. The open-nozzle fuel valve offers but slight pressure resistance to the pump discharge; consequently the pumping head consists of merely the pipe and check valve resistances, which are negligible. This fact enables the fuel pump to be designed with direct control of the pump plunger, with but slight reactions on the governor.

Types of Fuel Pumps.—Fuel pumps may be separated into three types or classes. The first of these covers the pump designs wherein the stroke of the fuel pump plunger is changed to meet the load variations, in this way controlling the amount of fuel handled by the pump.

The second class includes those pumps in which the suction valve is held open during a portion of the pump plunger's delivery stroke. In this way part of the fuel drawn into the pump plunger cavity is allowed to flow back through the suction valve. Only the oil handled by the pump during the latter part of its pressure stroke, and after the suction valve has closed, is forced into the spray valve.

The third type or class includes those pumps having a separate by-pass valve, which by-passes part of the oil handled by the pump back into the suction line.

American Diesel.—The numerous American engines that are still in service are equipped with the Bagtrup governor and fuel pump, appearing in Fig. 231. The mechanism consists of a pump body, in which reciprocates the plunger, one plunger for each engine cylinder, the plunger being driven by an eccentric mounted on the pump shaft. This shaft carries a gear which is actuated by a train of gears from the engine crankshaft. The suction valve is mechanically operated, by means of a bell-crank *B* and reach-rod *A*, from the plunger eccentric strap. The bell-crank fulcrums on an eccentric shaft *C* controlled by the governor sleeve. The suction valve is opened during the suction stroke and part of the discharge stroke. If the load is heavy the fulcrum pivot is raised, allowing the suction valve to close early; the fuel then is forced out through the discharge valve into the fuel injection valve. On light loads the governor lowers the fulcrum causing the suction valve to remain open for a greater

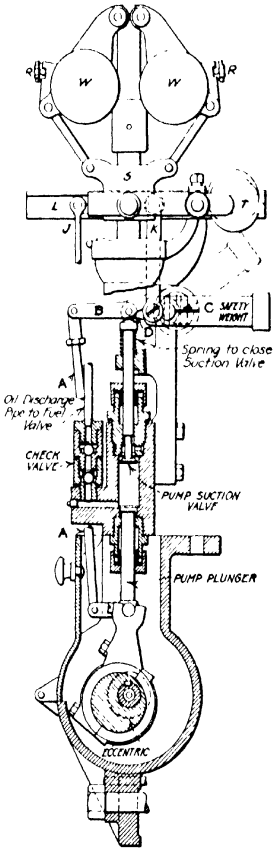


Fig. 231.—American Diesel fuel pump.

part of the plunger stroke; the oil then passes back through the suction valve.

The reaction of the governor sleeve is heavy, and the speed regulation is not as close as is demanded in electric plants. In starting the engine a hand crank is provided which allows the fuel line and injection valve to be charged before the engine is turned over.

Adjustments. Setting Suction Valve.—In setting the suction valve the governor should be blocked to the mid-position and the particular pump plunger placed on the upper dead-center. The suction valve lever should then clear the suction valve stem by $\frac{1}{32}$ in. This clearance can be secured by altering the length of the lever reach-rod A, which has turnbuckle ends. The governor sleeve should then be raised to its highest position and if the valve lever is not holding the suction valve open a slight amount the reach-rod should be readjusted to make it do so.

Leaky Valves.—The suction valve, being of the poppet type, frequently leaks. The proper method of regrinding is to remove the valve and cage. Disassembling, the spring is removed and the valve coated with emery flour and vaseline and again placed in the cage. A nail thrust through the cotter opening makes a convenient handle with which to turn the valve.

Discharge Valves.—The discharge line has two ball valves. When the oil is clean and free from dirt, little trouble is experienced. The chief attention is given to the valve seats. These seats tend to wear rounded, making a poor seal. When the seat is in this condition, it should be reamed to the correct 45-deg. angle; in this work the reamer must not chatter, or the seat will not be oil-tight. The lift of the valve should not exceed $\frac{1}{16}$ in. If it is greater, the ball valve will be slow in seating and will allow part of the oil to flow back through the valve opening. With heavy oils the seating is so slow that a small helical spring becomes necessary.

Levers and Pins.—On these old engines the pump levers and pins are generally badly worn. An engineer, when this condition exists, should promptly ream the pin bearings to a larger diameter and turn up new pins to conform. For this work an expansion reamer is desirable in order to handle all the various size pin bearings.

Busch-Sulzer Type B Diesel Fuel Pump.—While the mechanical details differ, the Busch-Sulzer Type B Diesel's fuel pump follows the same principle as does the American engine just discussed. Figure 232 is a cross-section of the Type B pump, showing one plunger. This is along designs adopted by the majority of European and English Diesel builders. In this construction the plunger *A* is driven by an eccentric *B* keyed to the vertical governor shaft *C*. The suction valve is mechanically operated by a dog *F*, which swings on an eccentric *G*. This eccentric is mounted on a small shaft controlled by the governor *K* through a linkage *J* and bell-crank *I* shown. The suction valve plunger *D* is also driven by a fixed eccentric *E* on the vertical governor shaft, being 180 deg. behind the pump plunger eccentric. In Fig. 232 the pump plunger is at the end of the delivery stroke while the suction valve plunger is at the extreme inner position. As the governor shaft revolves, at one-half engine speed, the pump plunger moves to the end of its suction stroke; the suction plunger moves outward, lifting the suction valve off its seat; this allows the fuel to enter the pump cavity. As the pump plunger reverses and moves on its delivery stroke, the suction valve remains open, the oil flowing back into the suction line. At a stated point in the pump plunger travel, the suction plunger moves out of contact with the dog *F*; the valve now closes and the oil is forced out through the discharge value. If the load decreases, the rising governor sleeve shifts the center of the eccentric dog bearing. This allows the valve plunger to remain in

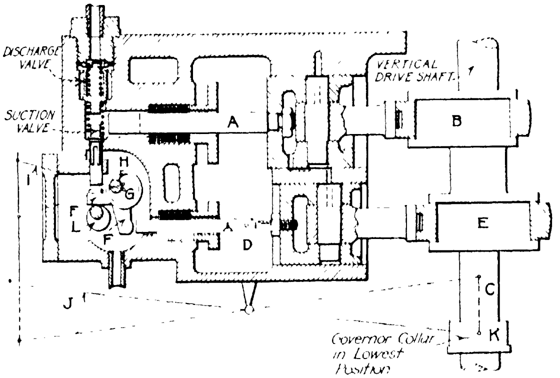


Fig. 232.—Busch-Sulzer Type B Diesel fuel pump.

contact with the suction valve for a longer interval, which permits more of the fuel charge to flow back into the suction line.

On these engines there is a pump plunger and suction valve plunger for each engine cylinder. It is possible to use only one suction valve plunger, but this individual suction valve mechanism offers opportunity for a closer adjustment of the functioning of each pump. On some of the four-cylinder engines the four plungers are arranged in a single row, while on other there are two sets of two plunger cavities each, placed end to end with the valve block in the center.

This fuel pump gives the closest possible speed regulation while the reaction on the governor is at a minimum. The resistance offered to the movement of the governor sleeve consists of the suction valve spring compression.

Filling the Fuel Pump.—To facilitate charging the pump and discharge line with oil, an eccentric shaft *L* is provided. This shaft is rotated by means of a hand lever, and, by so doing, the dog *F* is raised to its maximum lift. This lifts both the suction and discharge valves.

Stopping the Engine.—A less angular travel of the eccentric shaft *L* lifts only the suction valve. This relieves the pumps of the oil charge, and the engine stops from the lack of fuel.

Setting Pump Valves.—The four plunger pumps deliver the fuel to the four cylinders on all four cycles, *i.e.*, suction, compression, expansion and exhaust strokes. In timing the pump and the suction valve opening, the engine is slowly turned over, and the inner and outer dead-centers of the pump plunger eccentric are marked on the plunger. The engine is then turned over until the pump plunger is 7/16 in. from its discharge dead-center. The suction valve plunger, or regulating plunger, has at this point moved away from the dog or bell-crank, which leaves its contact with the suction valve stem. The suction valve plunger should be adjusted to give a clearance of 0.002 in. between the dog and valve stem when the pump plunger is in the position mentioned. In setting the valve, the governor collar must be central. In a two-plunger pump this means that the collar must be in the mid-point of its travel, while with a four-plunger pump the collar must be on its bottom position; to obtain the latter the governor springs must be removed.

Pump Valves.—The engineer need give but slight attention to the pump other than to the valves. Both the suction and discharge valves wear rapidly, as may be expected of any type of pump valve when dirty oil is handled. In regrinding, emery flour should be used, placing a very small quantity on the valve face. In some engines the discharge valve spring continually breaks. It is hard to determine the cause; at times a lighter spring relieves the trouble while on other pumps a heavier spring is required.

Type C Fuel Pump.—The pump used on the Busch-Sulzer Type C two-stroke-cycle Diesel is of the same general design as the Type B pump, although different in details.

McIntosh & Seymour Diesel Full Pump. The first Diesels manufactured by the McIntosh & Seymour Corporation were equipped with fuel pumps somewhat after the design appearing in Fig. 232. The engines were four-cylinder units, and the fuel pumps had two pumping plungers, one plunger for each pair of cylinders. The fuel from one plunger cavity passed through the discharge valve and pipe line into a block, called the distributor. This block contains two passages connected to the fuel lines leading to the two fuel injection valves. The cross-sections of these passages are controlled by needle valves. The fuel, entering the distributor, divides into two streams; the needle valves allow the operator to properly proportion the two oil streams.

The operating difficulty of this pumping system lies in the inability of the engineer to regulate the distribution of the fuel on varying loads. A setting of the needles that is correct for fuel load will not give the proper regulation at low loads since the resistances of the passages vary, due to a smaller quantity of oil entering the distributor. Another factor that prevents proper proportioning of the fuel is the partial clogging of one line; this throws almost all the oil into one cylinder.

With this form of pumping mechanism it is imperative that the oil is filtered to prevent the clogging of a distributor. Furthermore, the fuel valves must be kept in perfect condition since the smallest leak in a fuel injection valve lowers the resistance of this particular fuel line, allowing this cylinder to receive too large a proportion of the fuel from the pump.

The fuel pump, mentioned above, gave way to a design which is shown in Figs. 233 and 234. Unlike the distributor type pump which followed Sulzer Bros. patents, Fig. 233 is of exclusive American design. This pumping apparatus consists of two pumps, set at right angles, each being an outside packed double-plunger pump. The eccentric strap *E* driving the two

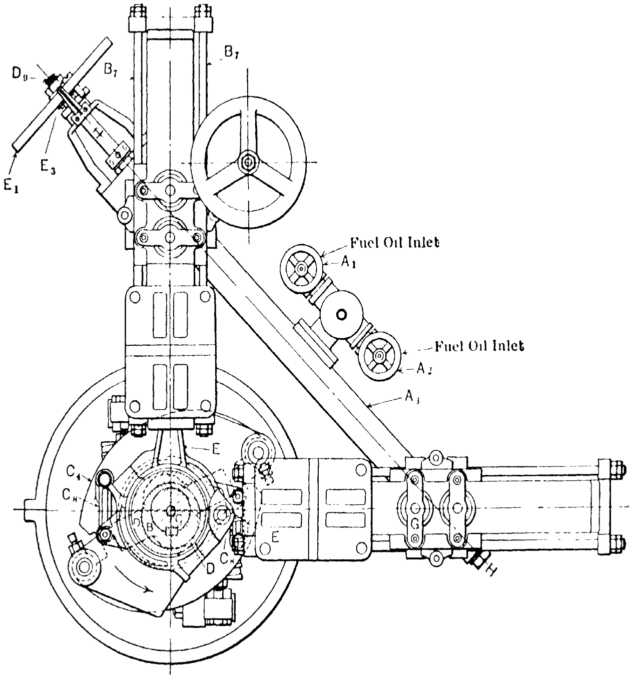


FIG. 233.—Governor and fuel pump, top view.

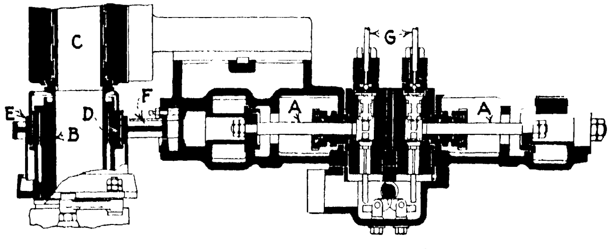


FIG. 234.—McIntosh & Seymour fuel pump.

plungers is mounted on an eccentric *D*, and the governor acts directly on the pump plungers, in this way regulating the amount of fuel pumped by varying the plunger stroke. The reaction on the governor of direct plunger-controlled pumps is considerable. McIntosh & Seymour partially avoid this by using two eccentrics; the eccentric *D*, controlled by the governor, drives the pump and is mounted on a second eccentric *B* that is keyed to the vertical governor shaft *C*. It is apparent that this offers a more accurate regulation of the pump stroke and a greater reduction in the reaction on the governor than can be secured by a single eccentric.

The pump suction valves are located below the discharge valves, being removed through the discharge valve opening. The latter valves are accessible by the removal of the valve cap or plug *G*. Below the suction valves of the two pumps is placed a shaft *H*, which has two milled cams, as shown in the cross-section. During the functioning of the pump the suction valve stems clear the shaft by means of the depressions in the shaft. Rotation of the shaft lifts the suction valve, thereby filling the pump with oil.

Pump Valves.—After continued regrindings of the suction valve, the stem may touch the cam at its lowest position. This cocks open the valve, preventing any fuel reaching the injection valve. Grinding the end of the stem will allow it to clear the cam. This same trouble frequently occurs in regrinding the discharge valve, causing it to strike the suction valve. The clearance between the two should be approximately $^5\!/_{16}$ in.

The valve springs at times break. This is probably due to fatigue; replacement by a spring of smaller wire in most cases seems to remedy this trouble. The discharge valve should be allowed a lift of at least 0.025 in. The valves have a 60-deg. slope. In regrinding more care is necessary than with a 45-deg. valve seat.

The discharge-valve cap is sealed with a metallic gasket. This gasket must be absolutely clean to avoid leaks. Leaks around the valve cap are inexcusable in pumps; not only is the dripping oil unsightly but it also impairs the engine regulation.

The plunger stuffing-box glands are best packed with shredded lead or vulcanbestos. Watch engineers should be cautioned not to tighten up on the glands with too great a pressure. This binds the pump plunger, scoring it and increasing the governor reactions.

Fuel Pump Timing.—The fuel pump and eccentrics come from the factory properly timed, and no change is necessary. In case it appears that the eccentric has slipped, the best method of checking the setting is to place the engine with the piston of No. 1 cylinder on bottom dead-center, just starting the compression stroke; in this position the pump plunger should be at the end of its delivery stroke.

McIntosh & Seymour Marine Diesel Fuel Pump.—The Four-stroke-cycle trunk piston Marine Diesel of this make is equipped with a fuel pump along somewhat similar lines. In this pump, Fig. 235, six plungers, placed horizontally, are operated by eccentrics mounted on eccentrics. This second set of eccentrics is keyed to the pump shaft, which is moved lengthwise by means of the manual control lever. These eccentrics are made with their axes at angles with the eccentric shaft center line. Shifting the shaft lengthwise increases or decreases the travel of the pump plungers; the travel of the plungers can be completely cut out by reducing the eccentric throw to zero. The longitudinal movement of this pump shaft is accomplished by the control lever through the air-starting control shaft, as has been discussed in Chapter VIII.

As a precaution against overspeeding, a governor is mounted on the pump shaft in the extension of the pump and operates, through a linkage, a small shaft *A*, which raises the suction valves. In order to allow a single cylinder to be cut out at will, a hand-lifter is provided for each suction valve. Another feature of excellence is the glass cup on the fuel suction line. This shows when the line is empty. To prime the engine when the engine is stopped, a handwheel is placed on the pump shaft, a few turns of which fills the suction line and pump cavities.

Fuel Pump Timing.—For each cylinder, the pump plunger must be at the end of its delivery stroke when the piston for that particular cylinder is at bottom dead-center, just beginning the compression stroke. The linkage to the control lever must be adjusted to give the pump plungers zero travel when the control lever is at both the start and stop positions.

The six-cylinder box frame Diesel of this make has a fuel pump that in principle is the same as the one shown in Fig. 233, but differs in construction. Each cylinder has a separate fuel pump; the six pumps are set on a ring around the governor shaft and the plungers are driven from the eccentric by connecting rods. The suction valve is raised by moving a wedge ring so that the valve stems are raised.

The crosshead type marine Diesels have a fuel pump of a design similar to Fig. 235; the large stationary engines have pumps

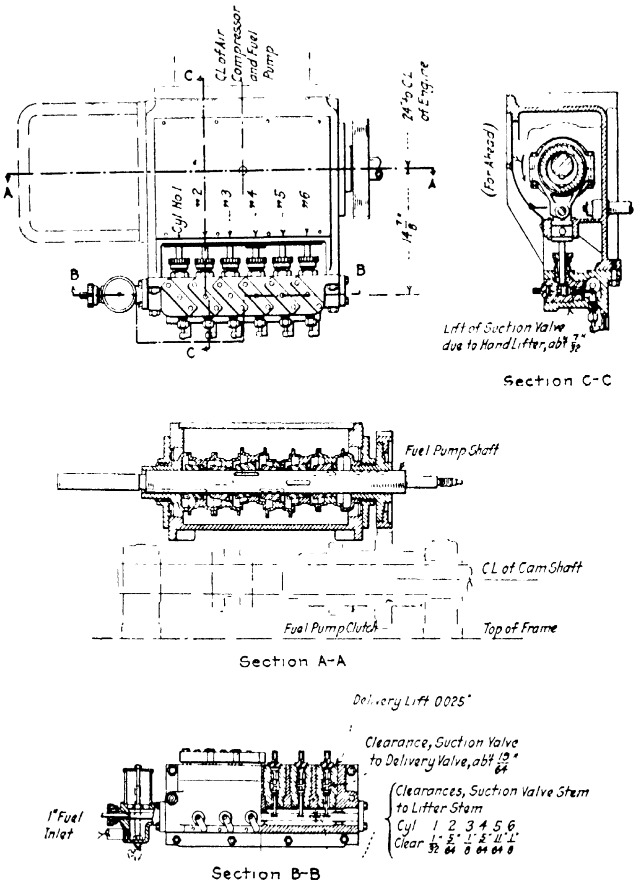


FIG. 235.—McIntosh & Seymour marine four-stroke-cycle Diesel

similar to Fig. 233 but these are mounted on the base close to the camshaft and the governor is on this shaft.

McEwen Diesel Fuel Pump.—The McEwen Diesel has an open-nozzle fuel valve; consequently the pump is a departure from the designs already discussed. As will be noted in Fig.

236, the pump, for a single-cylinder engine, consists of a plunger housing, to which is bolted the valve body, and the pump plunger with actuating mechanism. In action the cam, through the rocker *D*, moves the pump plunger *E* to the left, which action forces the fuel charge out through the discharge line to the fuel injection valve. As the cam in turning releases the thrust on the rocker, the plunger spring forces the plunger to the right; this draws in a charge of oil through the suction valve. The pump plunger on this stroke moves to the right until it strikes the governor wedge *C*. The length of the plunger stroke determines the amount of oil entering the cylinder. The governor is linked to the wedge *C* and, as the engine speeds up, shoves the wedge farther in, decreasing the distance from the wedge to the face of the plunger slot. This reduces the plunger stroke.

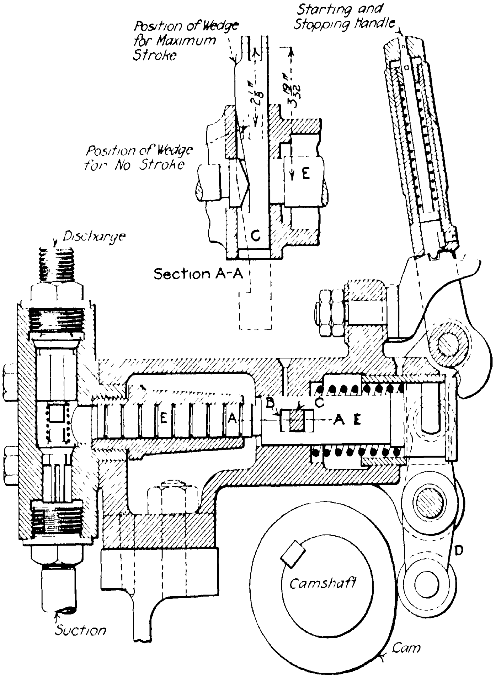

Starting and Stopping Handle
Position of Wedge for Maximum Stroke
A
Position of Wedge for No Stroke
E
C
Discharge
Section A-A
E
A
B
C
A E
D
Camshaft
Cam
Suction

FIG. 236.—McEwen fuel pump.

This pump is one of the simplest in use and has the advantage, as have all wedge-governored pumps, of offering but slight resistance to any governor movement. The reaction on the wedge is merely that due to the plunger spring tension when the wedge is in contact with the plunger. This is slight and occurs only at the point of extreme suction travel of the plunger. As a consequence, the governor can be light and extremely sensitive. This is of distinct advantage in alternating-current work, or in any other work necessitating close regulation.

Pump Valves.—Both suction and discharge valves are of the poppet wing type with seats having a 60-deg. slope. In regrinding these valves, the discharge valve must not be lowered enough to prevent a proper lift of the suction valve; this should be at least 0.03 in., while the discharge lift works best with a lift of around $\frac{5}{64}$ in.

Pump Plunger.—The plunger has no stuffing-box, being provided with sealing grooves. Since the pumping head is very low, but little leakage will occur even though the pump sleeve is considerably worn.

Pump Timing.—The pump cam is keyed to the engine layshaft and requires no alteration in timing. However, the pump plunger should be at the end of its discharge stroke when the engine piston is on out dead-center, just starting the compression stroke.

In setting the governor wedge for no-load conditions, after the weights are thrown out to their greatest travel, the wedge should be moved in until the slot is in contact with the thickest part of the wedge. Then after throwing the pump plunger to the end of its discharge by turning the engine over, there should be not more than 0.01-in. play between the wedge and the inner edge of the slot. This is the position of the wedge when no fuel is pumped. For full-load condition, at which event the weight arms are at their maximum position, the wedge should have moved in the slot so that the slot edge just strikes the wedge at the point where the wedge slope begins. This point should be $2\frac{1}{8}$ in. from this first or no-load position.

Snow Diesel Fuel Pump.—The Snow horizontal Diesel engine has a fuel pump designed with wedge control. This pump, which appears in Fig. 237, has the plunger in two sections. The driving end of this plunger is hollow and carries the slot for the wedge. It is also provided with a pin which bears against the cam lever roller at all times, being held by a compressed spring. With this design, on short plunger strokes, as on low load, the roller does not strike the plunger end when traveling at maximum speed. Instead the plunger pin moves in, being resisted by the spring, until the spring compression overcomes the pump plunger resistance. At this point the pump plunger starts on the delivery stroke. The result is a quiet pump with a minimum of wear and shocks on the mechanism. The plunger wedge is fastened to the reach-rod *A*, which is under control of the governor.

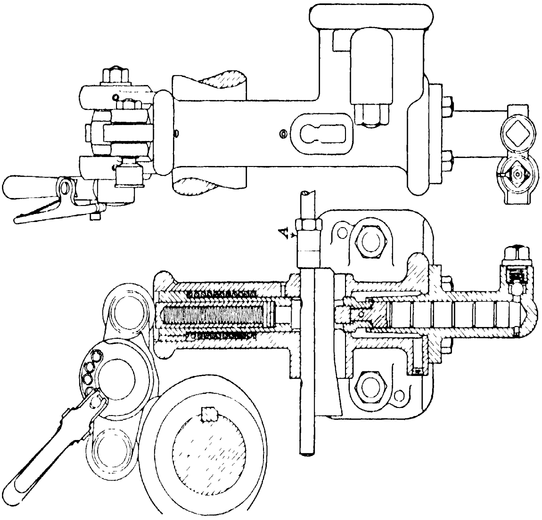


Fig. 237.—Snow, horizontal Diesel fuel pump.

Pump Valves—The pump valves are of the ball type. In case leaks develop, a hardwood stick can be placed on the ball and struck a light blow with a hammer. This will give the ball a new seat on the housing. In event the ball is scored or rough, the only remedy is replacement with a new ball; even here the ball must be given a new seat in the housing. The ball valve, if it has any great amount of lift, is sluggish in closing. A lift for a pump-ball valve ought never to exceed 1/16 in. If more than this, a light coil spring should be inserted above the ball to quicken its action.

Timing the Pump.—In adjusting the pump plunger for no-load and full-load strokes, the reach-rod *A* should be adjusted to allow the plunger to touch the wedge at the beginning of the wedge slope, when the governor is at its lowest position, which is the full-load condition. For no load, when the governor is blocked open, the wedge should have moved to a position where

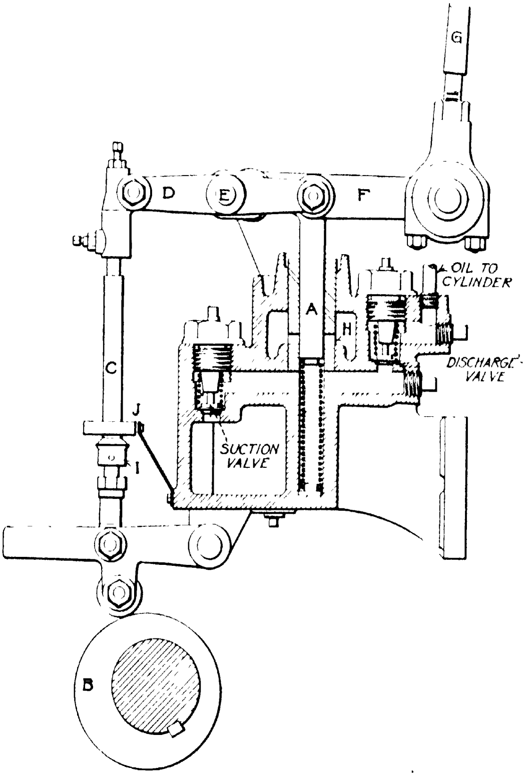


FIG. 238.—Allis-Chalmers fuel pump.

the plunger has a slight clearance between the slot edge and the wedge, on the inner position of the pump plunger.

Allis-Chalmers Diesel Fuel Pump.—The fuel pump used on the Allis-Chalmers engine is shown in Fig. 238. The pump plunger is actuated, through a rock-shaft *D* and the connection rod *C*, by the fuel cam below the pump. The plunger rock-shaft *D* is fulcrumed at *E* on the lever *F*, which is under control of the governor through the reach-rod *G*. As the engine speeds up, the reach-rod moves downward; this raises the fulcrum pivot *E*. This motion of the fulcrum raises the pump plunger *A* until its end is above the by-pass slot *H* at the end of the suction stroke since the plunger spring holds the rod *C* in contact with the cam *B*. As the cam rotates, the plunger receives a constant stroke, but, since part of the stroke occurs before the by-pass slot is covered by the plunger, the first portion of the oil displaced by the plunger flows through the slots *H*. As soon as the plunger passes the slots, the oil below the plunger is forced through the discharge valve to the atomizer. Since an open-nozzle fuel valve is employed in connection with the pump, the latter has but the pipe resistance to overcome. The governor, then, is called upon to oppose only a slight reaction and consequently can be sensitive without danger of hunting.

The cam connection rod *C* is provided with a special head having adjusting set-screws. Proper manipulation of these set-screws will alter the effective stroke of the pump plunger.

In stopping the engine the hand lever is raised, allowing the release band *J* to grip the lock *I* on the rod *C*. This prevents the rod from dropping into contact with the cam. In starting, the handle is used to prime the pump.

The pump is fitted with a plunger and rocker mechanism for each cylinder of engine, with individual cams on the layshaft.

Fulton Diesel Fuel Pump.—Although the Fulton Diesel has closed type fuel valves, the amount of fuel handled by the pump is altered by a change in the stroke of the pump plunger. However, no great reaction is thrown upon the governor since the pump stroke is changed during the suction period. This is shown in Fig. 239. The pump is a steel body containing a plunger cavity and check valves for each engine cylinder. The pump plungers *B* are moved by individual eccentrics *A*, which are mounted on the camshaft. The plungers are not fixed to the eccentric strap, but the latter carries a sleeve *C* which is fitted with a latch *D*. With the pump plunger in the position shown in Fig. 239, the rotation of the eccentric forces the plunger downward, delivering oil to the engine cylinder. The latch during this time rests under the collar *D*. As the eccentric stroke reverses the latch *D* by engaging the plunger collar lifts the pump plunger. The eccentric, latch and plunger then move upward until the latch roller *F* strikes the cam *G*, whereupon the latch block is disengaged from under the plunger collar *D*. After this the further upward travel of the eccentric strap *A* and crosshead *C* has no effect on the plunger, the latter remaining stationary. On the return stroke the crosshead *C* comes into contact with the plunger head and forces the plunger downward, forcing oil into the fuel or spray valve. The governor, through a rod or shaft *H*, and by shifting the cam position, causes the latch to be disengaged earlier or later as the load demands, thus allowing the pump-suction stroke to be long or short. The latch is quite similar in arrangement to the disengaging latch block of a Corliss valve gear.

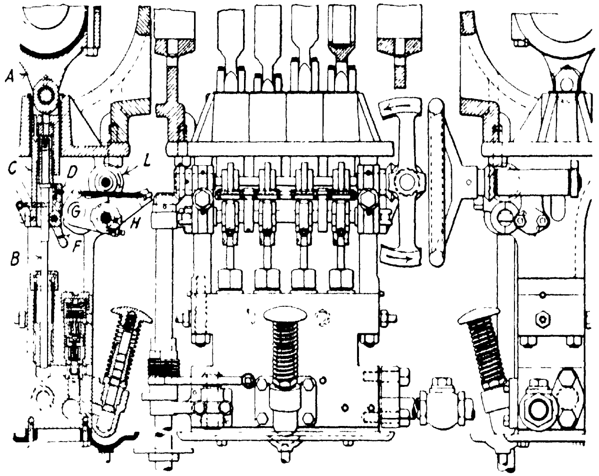


Fig. 239.—Fulton fuel pump.

The pump has two poppet sleeve valves on each discharge connection while one wing-type poppet valve is provided for the suction. A hand plunger is fitted to allow priming before starting. To stop the engine the hand wheel *J* or the lower hand wheel connected to the shaft *K* rotates the cam *L*, which in turn throws the cam *G* to its earliest position where the latch is completely out of contact with the plunger collar.

This is as accurate a measuring device as used on any Diesel, and the fact that the governoring must be done on the suction stroke of the pump plunger somewhat early for close regulation of the fuel into any particular cylinder, does not affect the speed control of the engine to any objectionable degree.

Cramp's B. & W. Diesel Fuel Pump.—The fuel pump of the Burmeister and Wain Marine Diesel is shown in Fig. 240. In this the pump suction valve lift is altered by changing the position of the eccentric shaft upon which the lifting dog rests. Each engine cylinder is provided with a separate fuel pump.

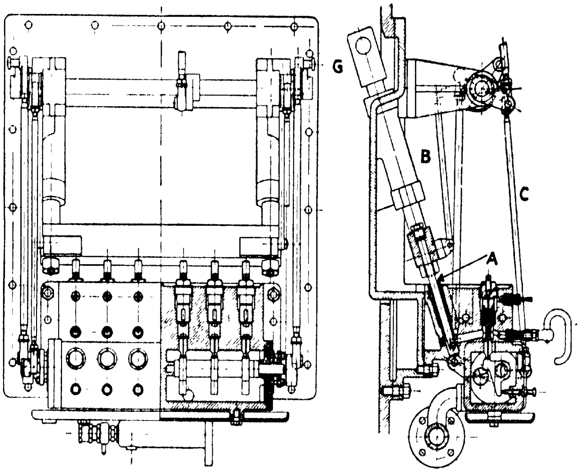


FIG. 240.—Cramp's B. & W. fuel pump.

The pump plunger *A* is driven by an eccentric on the engine camshaft. The link *B* through a bell crank and the rod *C* opens and closes the suction valve by means of the valve dog as shown.

De La Vergne FD Diesel Fuel Pump.—Figure 241 is an external view of the fuel pump used on the FD engine. The governor acts upon a by-pass valve *a* through a system of levers. The fuel charges are controlled by the movement of this by-pass valve.

Pacific Diesel Fuel Pump.—The Pacific Diesel has the fuel pump shown in Fig. 241 *A*. The action is self evident.

Nelseco Marine Diesel Fuel Pump.—The fuel pump fitted to the Nelseco Marine Diesel appears in Fig. 242. The engine speed can be altered by the control lever shown, through a linkage which alters the position of the governor lever fulcrum. The governor then maintains this desired speed by changing the period of suction valve opening. The fuel pump consists of the

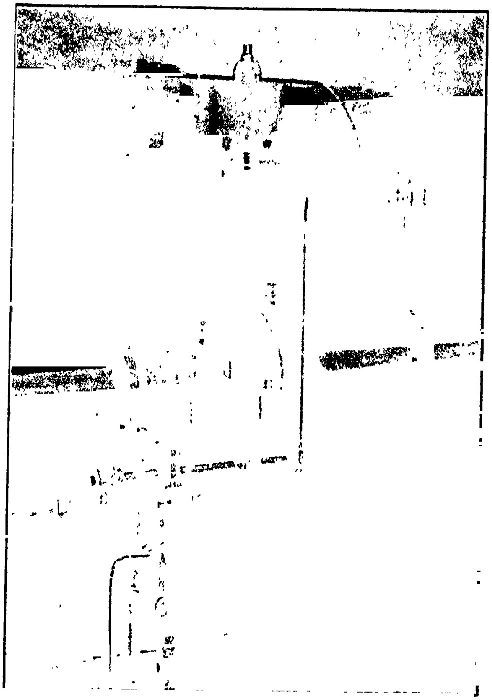


FIG. 241.—De La Vergne Type FD fuel pump.

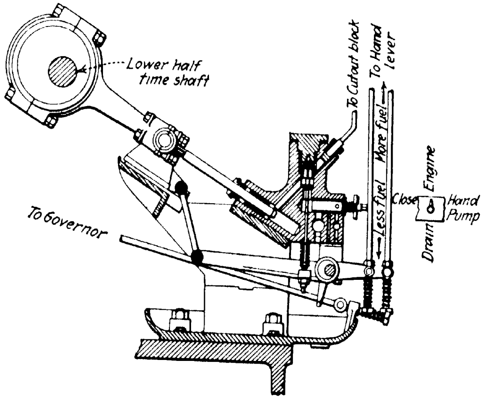


FIG. 241*A*.—Pacific fuel pump.

working plunger driven by an eccentric, and the suction and discharge valves. The suction valve is held open through

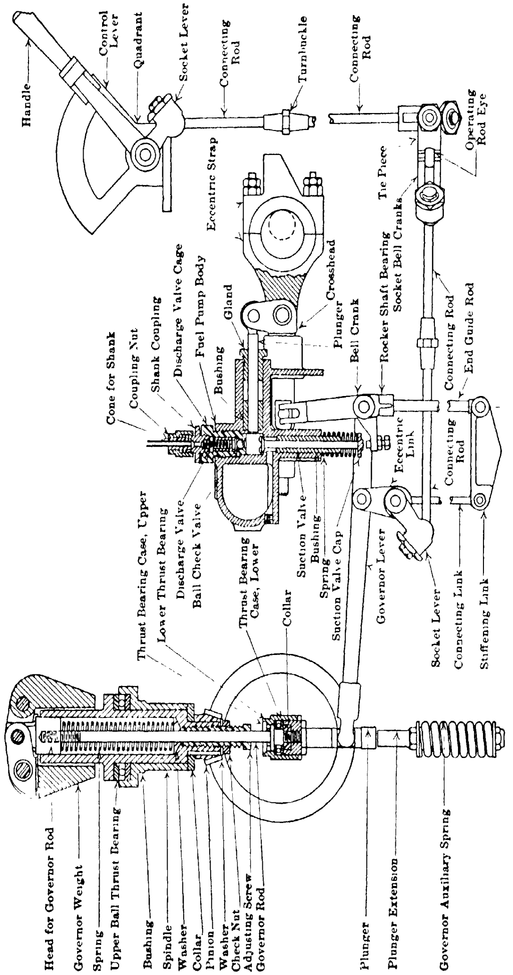


FIG. 242.—Nelseco marine Diesel fuel pump.

part of the delivery stroke of the pump, the length of this period being under governor control.

Timing Fuel Pump.—To adjust the pump proceed as follows: Set the fuel control lever at division No. 1 and lock. Bar the engine over until some two plungers are at the inner end of their stroke. Then adjust the adjusting screw on the corresponding suction valves until a piece of paper placed between the adjusting screw point and the fuel suction valve end is just barely held. Adjust all the other suction valves in the same way, still keeping the fuel control handle locked in the same place.

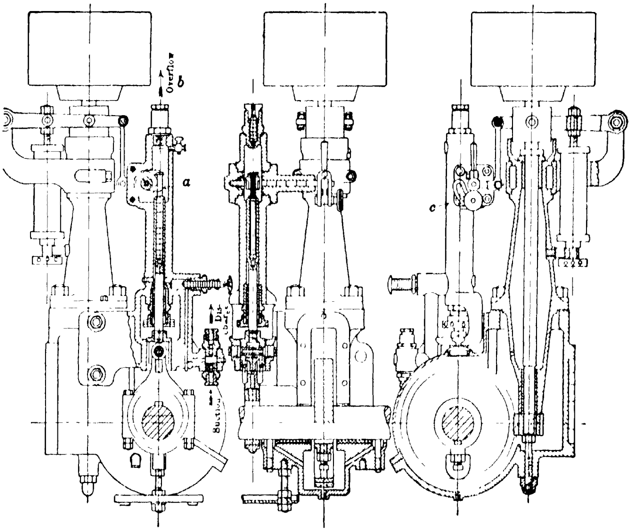


Fig 243—National Transit fuel pump, old model.

National Transit Diesel Fuel Pump.—The fuel pump on the first National Transit engines has a differential plunger, the upper end being hollow and provided with a cut-off valve, Fig. 243. This differential plunger is driven by an eccentric keyed to the camshaft. The cut-off valve is under control of the governor through a link and bell-crank. The pump functions as follows: On full load the governor sleeve is as shown in the figure; this position is such that the cut-off valve leaves its seat on the plunger as soon as the latter starts on the downward or suction stroke. The oil is then drawn through the suction valve and fills the passage in the plunger and the chamber *A* above the differential plunger. On the upward or discharge stroke, the cut-off valve being open, the large end of the plunger displaces the oil in this chamber and forces it down through the plunger and out the discharge valve. On low loads the governor bell-crank moves downward; this allows the valve to remain seated on the plunger during part of the downward stroke. As soon as the plunger moves below this point, the valve rests on the bell-crank fingers and leaves its seat. The chamber fills with oil, and on the upward stroke the fuel is forced out the discharge valve until the plunger comes in contact with the cut-off valve. The cut-off valve then seats, and all the oil displaced by the further movement of the plunger passes out through the overflow valve *B* at the top of the pump body.

Pump Valves.—To insure close regulation, the cut-off valve seat must be in good condition. Even though it is rather unhandy to reach, at least monthly the valve should be removed and, if the seat is not smooth, it should be reground. To remove the valve, the overflow-pipe joint can be broken and the check-valve cage lifted.

Timing Fuel Pump.—The plunger eccentric is keyed to the camshaft, and no change of the timing is necessary.

In timing the cut-off valve, the valve should just start to leave the seat as the pump plunger commences its downward stroke when the governor is at rest and the collar in its lowest position. This is the full-load position. The governor springs should then be removed and the weights moved to their maximum outward positions, which raises the collar to its highest point. The cut-off valve should now be seated on the plunger during the entire stroke of the plunger, with the exception of about $\frac{5}{1000}$ in. This is the no-load position of the valve. An adjusting lever and lock *C* are placed on the bell-crank outside of the casing. This allows the timing of the valve to be altered to conform to the desired setting. To stop the engine this lever is moved to lower the cut-off valve.

National Transit 1918 Diesel.—The later National Transit Diesels are supplied with a fuel pump that, in form, resembles the pump discussed above but which functions on an entirely different principle. This pump appears in Fig. 244. In this design the pump plunger is actuated by a rocker and cam in place of the former's eccentric and strap. The plunger *P* is hollow and carries at its top the cut-off valve *V*. This valve has a triangular shank which extends down into the hollow plunger, and

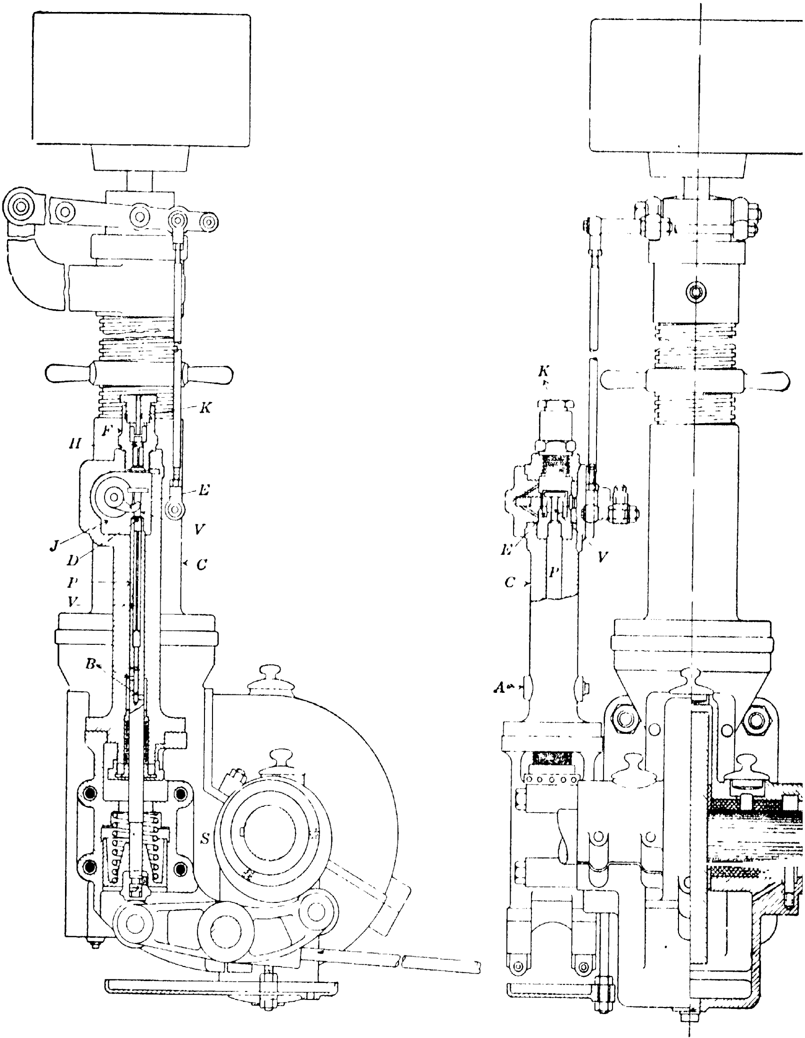


FIG. 244.—Fuel pump and governor, Type D-3, National Transit Diesel

the valve is controlled by the fingered or forked lever *E*. This lever is moved by the governor and limits the downward travel of the valve *V*. As the plunger is forced downward on its suction stroke by the spring *S*, the oil enters the pump body at the opening *A* and, passing through the ports *B*, enters the hollow plunger. Since the lever *E* has engaged the valve *V* at some point on the downward stroke of the plunger *P*, this valve is open and the oil fills the cavity *J*. On the upward stroke of the plunger the oil displaced flows back through the open cut-off valve *V*, through the ports *B* and suction opening *A*, to the source of supply. As the plunger continues its travel, it comes in contact with the valve *V*, which thereupon seats at *D*. The continued travel of the plunger entails a further displacement of oil, which lifts the discharge valve *H* and is forced to the fuel injection nozzle. It will be observed that, while with the pump in Fig. 145 on no load the cut-off valve was closed during the entire plunger stroke, with this pump the cut-off or suction valve *V* is opened during the entire plunger stroke when the governor is in the no-load position. This change in design eliminates the suction valve and the overflow valve while improving the fuel-measuring accuracy of the pump.

Standard Diesel Engine Fuel Pump.—The pump of this horizontal two-cycle Diesel engine is illustrated in Fig. 245. The fuel pump is a plain barrel *B*, having a plunger directly under the influence of a Rites inertia governor. The barrel and plunger *A* is the air injection control. Both the suction and discharge openings have two sets of wing poppet valves. The pump plunger is pivoted to a bell-crank which, in turn, is driven by the eccentric rod. The eccentric-rod side of the crank has a worm-screw adjustment whereby the stroke of the pump, for any given position of the governor weight arm, may be altered. Lowering the rod end gives a greater stroke to the pump plunger, and raising it, a lessened stroke. In timing the engine a good practical plan to set the pump stroke for no-load conditions is to first lower the eccentric until the engine speeds up above normal. Then the eccentric rod should be raised by the screw until, at a slight overspeed, the plunger movement is zero. This position of the eccentric end should give sufficient stroke to the plunger at the full-load position of the governor.

In stopping the engine the lock screw *C* is loosened, cutting out the pump. Since the engine is two-cycle, the pump works against a much greater pressure than exists with a four-stroke-cycle engine, even though an open nozzle is employed on both engines. For this reason the wear on the pump valves is greater, and regrinding is more frequent. The engineer must give special attention to the plunger stuffing-box, as a small leakage effects the engine's regulation.

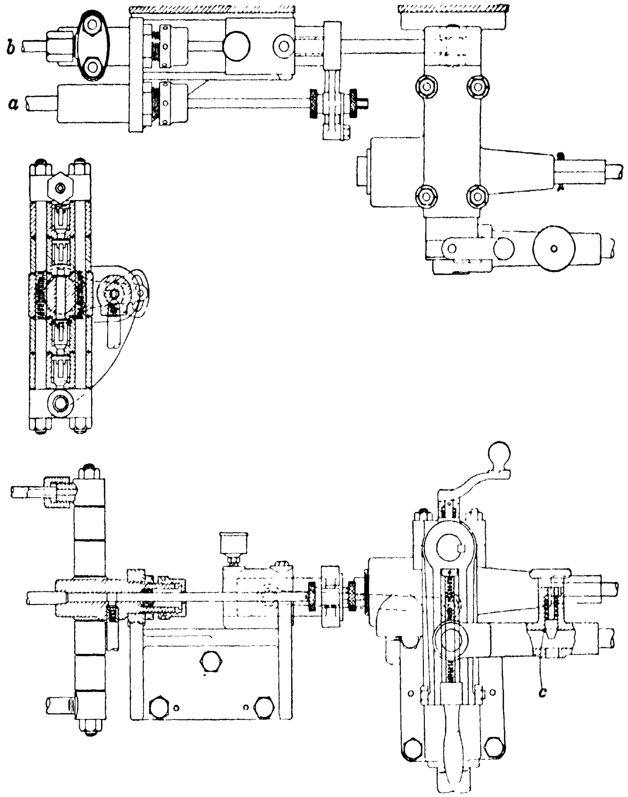


Fig. 245.—Standard horizontal-Diesel fuel pump.

The Standard Vertical Diesel Fuel Pump.—The fuel pump of this two-cylinder vertical engine is quite similar to the pump on the horizontal Standard engine. The governor which is a Rites inertia shaft governor controls the length of the pump strokes by altering the throw of the shaft eccentric through the rocker *A* and reach rod *B*. The rod *B* is connected to a bell crank which is mounted on a short shaft carrying a small crank *D* at its other end. This crank *D* fits into a slot in the plunger crosshead as shown by the side illustration in Fig. 246. Each engine cylinder has its individual fuel pump. Both suction and discharge of the pump are provided with two valves.

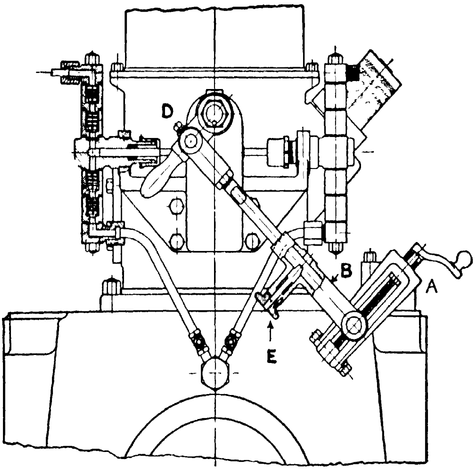


FIG. 246.—Standard vertical engine pump.

A hand crank on the rocker *A* allows a change in the pump stroke at any given eccentric throw, thereby providing a wide-speed range. To stop the engine the detent *E* is unfastened which disengages the reachrod and so causes the pump action to cease.

In this design the pump is reduced to the minimum number of parts. The governor weight is such that the reaction of the pump shows no effect on the speed regulation.

Winton Diesel Fuel Pump.—This fuel pump Fig. 247 is radically different from all other well-known pumps. The fuel pump plunger *B* is driven by the eccentric *A*. Linked to the plunger yoke is a system of levers *C* moving a crank *D*, which in turn gives the suction control plunger *F* a reciprocating motion. The crank *D* is mounted upon an eccentric sleeve *E* which is controlled by the governor.

The upward or suction stroke of the plunger *B* causes the suction plunger *F* to uncover the port *S* whereupon oil enters the pump cavity around the ball-check valve, from the suction line *T*. On the downward pump plunger stroke the oil in the plunger cavity is forced into the engine fuel valve. The suction or measuring plunger moves to the left, covering the oil port. If the load drops, the increased speed causes the governor to shift the position of the eccentric *E*. Under these conditions the measuring plunger *F* does not move as far to the right and returns, covering the oil port, earlier in the pump plunger suction stroke. A smaller amount of oil is allowed to enter port *S*. At the extreme high governor position the plunger *F* fails to uncover the port *S*, thus entirely cutting off the oil to the fuel pump.

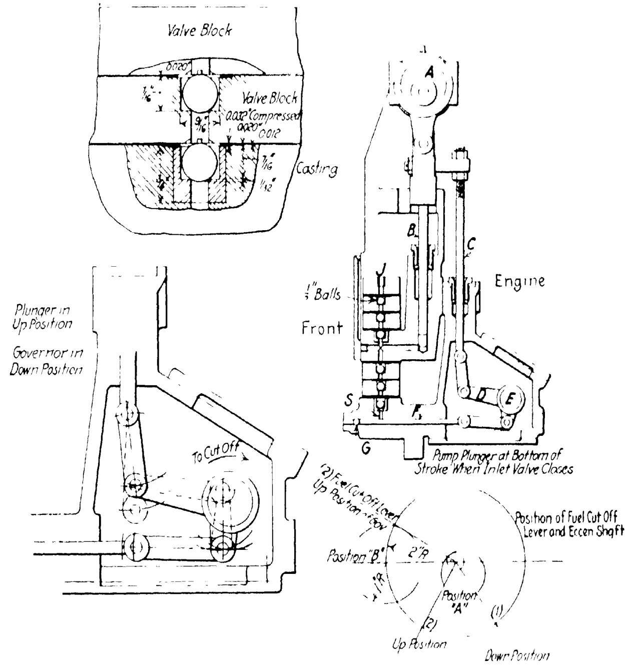


FIG. 217.—Winton fuel pump.

In Fig. 247 *B* is shown the position of the plunger *F* for full-load conditions at the time the pump plunger is at the top of its stroke. It will be noticed that at this load the plunger *F* fails to cover the port *S*. At no load the plunger *F* is in the position shown in dotted outline; here the port *S* is not uncovered at all.

Nordberg Type VE Diesel Fuel Pump.—The VE Diesel has the fuel pumps illustrated in Fig. 248. It consists of a pump body carrying suction valve *A* and discharge valve *B* and a plunger *C*.

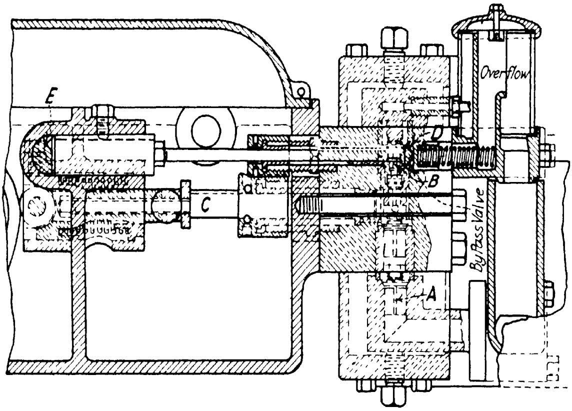


Fig. 248.—Nordberg VE pump.

The plunger is driven by a cam mounted in a layshaft along the engine frame. The plunger makes a constant stroke, handling a fixed amount of oil regardless of load. The quantity of fuel passing out through the pump discharge valve is regulated by the by-pass valve *D*. This by-pass valve is driven by a train of gears located at the governor, acting on the cam *E*. The governor, as shown in Fig. 249, by means of a reach-rod controls an idle mitre pinion meshing with gears on the by-pass camshaft. This shaft is in two pieces, the two ends having mitre gears meshing with the governor controlled gear shown in Fig. 250. A change in the position of the governor sleeve alters the position of the governor mitre gear. The shifting of this gear causes that part of the by-pass valve shaft that carries the by-pass valve cam to move ahead or behind the portion of the shaft that is geared to the engine crankshaft. This causes the by-pass valve cam to close the by-pass earlier or later as the case may be. The amount of oil by-passed back into the suction is thus under control and so regulates the quantity actually forced to the fuel valve.

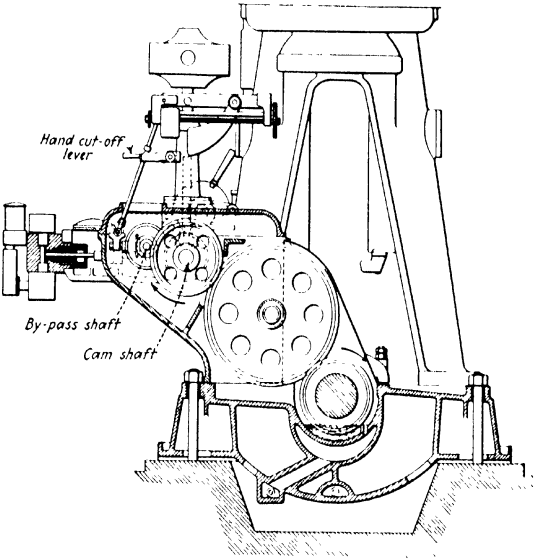


FIG. 249.—Governor and pump gear.

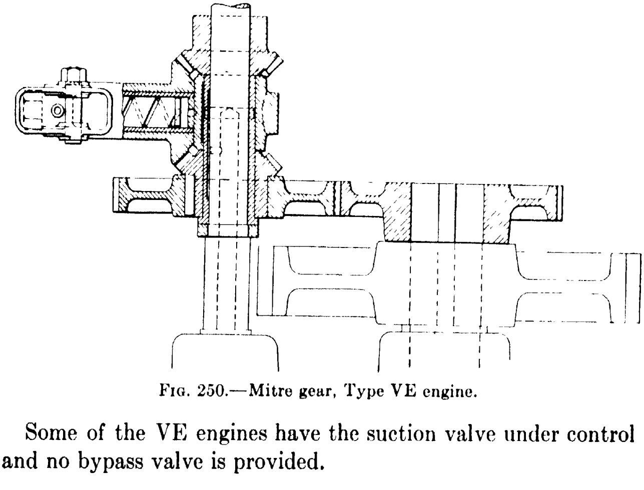


FIG. 250.—Mitre gear, Type VE engine.

Some of the VE engines have the suction valve under control and no bypass valve is provided.

Nordberg Type FD Fuel Pump.—The FD pump is shown in Fig. 251. The pump is a double unit; that is, it is provided with two plunger and by-pass valves so that two engine cylinders are supplied with oil from each unit. The pump plunger *A* is a hardened steel rod whose outer end is held in contact with the fuel cam on the camshaft by a spring inside the plunger housing. The regulation is obtained through the action of the by-pass valve. The valve rod of this valve is fitted with a scotch yoke *B*. A link connects this yoke to an eccentric mounted on the camshaft. The governor is connected to the fulcrum shaft of this link and, by shifting this eccentric fulcrum point alters the stroke of the by-pass valve. A hand pump *D* enables the oil lines to be primed before starting the engine.

FIG. 251.—Nordberg FD fuel pump.

Later FD Diesels have pumps similar to the type VE pumps.

Lombard Fuel Pump.—This pump is rather unusual, consisting as it does of two distinct pumps per cylinder. The first or primary pump works against no pressure and is merely a metering instrument. The stroke of the plunger is controlled by the governor and consequently the amount of oil drawn into this pump and then discharged into the secondary pump is regulated by the position of the governor sleeve. The secondary pump receives the metered charge and delivers it, against the injection air pressure, to the spray valve. As a result of this division of the pumping into a metering action and a high-pressure pumping action the governor is not called upon to overcome any great reaction from the pump. The regulation may, then, be very close.

General.—After the valves of any design of fuel pump have been replaced, after grinding, etc., the discharge line should be disconnected from the injection valve, if no by-pass valve is attached. The pump should then be primed. This will effect the escape of any air that might be trapped in the pump cylinder. In case the engine, on starting, fails to fire on any particular cylinder, the pump is probably air-bound, and the lines must be freed before oil will enter the fuel valve. At times the fuel-line check valve leaks, allowing air to flow down the pipe to the fuel pump. This will prevent the pump from delivering any oil.

Valve Grinding.—In grinding any poppet-type valve powdered glass, or emery flour, and vaseline make the best compound. The operator must exercise judgment in the amount of pressure exerted on the valve as it is rotated. If the pressure is excessive, the compound will groove the valve faces.

When a discharge valve or suction valve is under suspicion, the best method of ascertaining if it actually leaks is to disconnect the discharge line and place on the coupling a high-pressure gage. The engine can be turned over until the pump discharge pressure registers the usual value, 900 lb. with closed nozzle and about 200 lb. with open nozzle. The engine should then be stopped and the discharge pressure noted. If it falls, it can be taken as an indication of a leaky valve.

CHAPTER XV

GOVERNORS

TYPES. ADJUSTMENTS

Governors.—When an engine is carrying a constant load, the fuel pump delivers to the engine cylinders just enough fuel to enable the engine to overcome the resistance at the flywheel and the frictional resistance of the engine itself. This condition never exists in a power plant. The load may decrease, with the result that the fuel injected into the engine cylinder is greater than is necessary to overcome the lessened load. The engine would then speed up and wreck itself. If the load increased, the fuel charge would be insufficient and the engine would slow down and finally cease turning. Some form of fuel control is then imperative. On the marine Diesel, when the engineer is in constant attention upon the unit, a manual control is provided. This consists of some form of leverage that alters the pump stroke or the period of pump suction valve opening. Even with the marine engine an automatic overspeed device is usual to prevent disaster in event of non-attention on the part of the operator. The stationary engine receives no such attention since the engineer has many other duties to perform. Furthermore, a speed variation permissible on a marine engine cannot be allowed in an engine pulling either a central station or an industrial load. To obtain the speed regulation that such loads impose the device called a governor is used.

The close regulation of an engine's speed requires that the governor must be sensitive to speed changes. To attain this object the governor should possess a minimum internal frictional resistance, or otherwise too great a speed change will be reached before the governor overcomes its resistance and moves the control lever. It is clear that, without any internal resistance, the governor would respond to an infinitesimal change in engine speed with the result that the speed would be constant, within the power of observation. This internal resistance produces a sluggishness in the governor action which can be given a numerical value by defining it as the proportion which the change in speed necessary to produce the required governor movement bears to the total speed. That is, if N_1 is the speed at which the engine has been turning and N_2 is the speed the engine attains, at a change in load, before the governor acts upon the fuel pump then

$$\frac{N_1 - N_2}{N_1}$$

can be termed the sluggishness of the governor. For many kinds of work this value must be as small as possible; a value of $\frac{1}{100}$ is satisfactory in service. To secure this value the governor must be free from excessive friction; in other words, it must be sensitive. This, in many forms of governors, produces a "hunting" effect wherein the governor is constantly changing the position of the weights in order to meet the engine speed changes. Since the governor movement always follows the change in speed, if the governor is sensitive it may over-travel in meeting a speed increase; this over-travel will cause the engine's revolutions to decrease below normal; the latter then produces a governor movement which results in another increase in speed above normal. It is necessary that the governor sleeve has the same lifting force in every position for a given change of speed. Furthermore, the engine flywheel must be of ample size to absorb the speed fluctuations occurring during one cycle of the engine.

To avoid the "hunting" effect mentioned above, some governors are fitted with one of many designs of dashpots. Some of these dashpots consist of oil cylinders containing a piston connected to the governor-sleeve lever, having a passage between the two ends of the cylinder through which the oil can pass. This provides a resistance to prevent a governor movement due to any sudden speed change. The oil flows through the opening, thus allowing the governor to cope with any permanent change in engine speed. This, of course, makes the engine slow in action.

While the desirable governor must not be sluggish, it does not follow that a sensitive governor holds the engine speed within narrow limits on a change from no load to full load unless a correct design of pump is chosen. Much depends upon the effort which must be exerted on the fuel-pump mechanism to bring it from its minimum pumping position to its maximum position. Since a large change in engine speed gives the governor a maximum force to exert on the pump, it follows that the greatest permissible fluctuation in engine speed should be adopted to give the governor a high work capacity. A 3 per cent fluctuation or 3 per cent variation above or below normal speed is quite satisfactory for oil-engine governing. It does not follow that this variation occurs on any save extreme load changes. In the load changes that ordinarily take place, the fluctuations should not exceed 1 per cent since the governor-sleeve travel is small on small load changes.

American Diesel Governor.—The governor of the American Diesel and the Busch-Sulzer Type A engines is of the fly-ball type. The two weight arms *W*, which are held by a spring at *RR*, Fig. 231, act upon a sleeve *S*, which in turn raises or lowers a lever *L*. This lever is connected by the link *K* to the fuel pump suction valve rocker shaft *C* and alters the period of valve opening, as described in the preceding chapter. To absorb small speed variations an oil dashpot, not shown, is linked to the governor-sleeve lever at *J*; a speed regulator is also included, this being in the form of a weighted arm as appears in the illustration at *T*. As a protection in event the governor breaks, a safety-weight arm is fitted to the fulcrum shaft. This weight will pull the shaft into the inoperative position if the governor links break. This governor is not a type that allows any great permanent change in the engine speed; 20 per cent is as much as can be obtained while maintaining the governor's stability.

Adjustments.—The governor shows a decided tendency to wear at the links and pins. About every two years it is necessary to reream the link-pin bearings and turn up new pins. These pins are best made of brass to allow the wear to occur on the pins rather than on the links. It is much easier to turn up new pins than it is to re-ream the links. For the latter purpose an expansion rereamer is more serviceable than is the ordinary kind. The blades can be extended, giving a greater capacity for reaming.

Jahns System.—The Jahns governor has been adopted by the builders of the Busch-Sulzer Type B Diesel, the Snow Diesel, and the National Transit Diesel.

The construction of the Jahns governor may be readily seen from Figs. 252 and 253. The two weights *AA* are guided in a radial straight line perpendicular to the spindle by three rolls

C on the lower surface sustaining the weights, and two rolls, not shown, on the sides resisting the force of inertia which would tend to keep the weights revolving at the same rate while the engine—and, therefore, the governor casing—increases or decreases its speed by an infinitesimal amount. The centrifugal force of each weight acts directly upon its spring so that all lever joints are entirely free from any centrifugal or spring force.

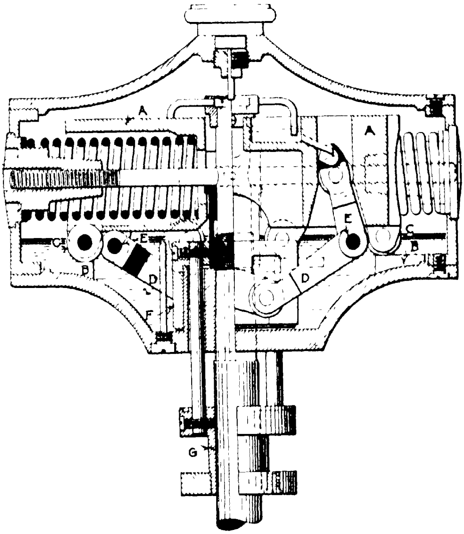


Fig. 252.—Jahns System governor.

The transmission of the motion of the weights to the sliding sleeve on the spindle is effected by the bell-cranks *D* fulcrumed on the lower casing, the upper arms engaging, by means of rolls, the vertical straight slots in the weights, while the lower arms engage sloping slots in the spindle sleeve *F*. The angle of this slope is fixed in such a manner that the centrifugal force of the weights, as transmitted to the sleeve, is a practically constant force, being the same for each position of the sleeve throughout its stroke. This collar *G* to which is connected the pump-control lever receives its motion from the spindle collar.

The Jahns governor casing is entirely enclosed, and all of the lower pins and slides are in a bath of oil. The oiling of the upper pins and slides is effected through the oil cup on the top of the governor. The oiling can be effected while the governor is in motion. When the oil has attained a certain height any additional amount will cause overflow. This surplus is conducted to the rubbing surfaces of the sliding spool located below and outside of the casing. In this manner every point of possible friction is automatically oiled.

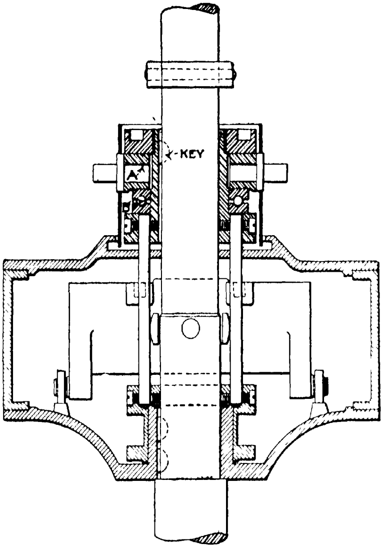


FIG. 253.—Busch-Sulzer Type B Jahns governor.

The Jahns governor possesses great sensitiveness owing to its extremely small internal friction and the fact that none of the working parts may become rusted or clogged with dust.

Owing to its low internal resistance, the Jahns governor has great capacity, the percentage of available energy for regulation as compared with the total energy developed by the weights being 99.8 per cent.

The Jahns governor, as supplied on the oil engine, always has a speed regulator attached. This regulator is usually made to give a variation of plus and minus 5 per cent of the standard speed. The arrangement of the governor and regulator varies slightly in each engine.

Busch-Sulzer Diesel Governor.—The Jahns governor fitted to the Busch-Sulzer engine appears in Fig. 253. The governor sleeve is placed above the governor case rather than below as is the standard practice. This brings the sleeve within range of the fuel pump. The governor acts on the suction valve of the fuel pump, as shown in Fig. 232. The arrangement of the governor and fuel pumps on the vertical shaft is seen in Fig. 254.

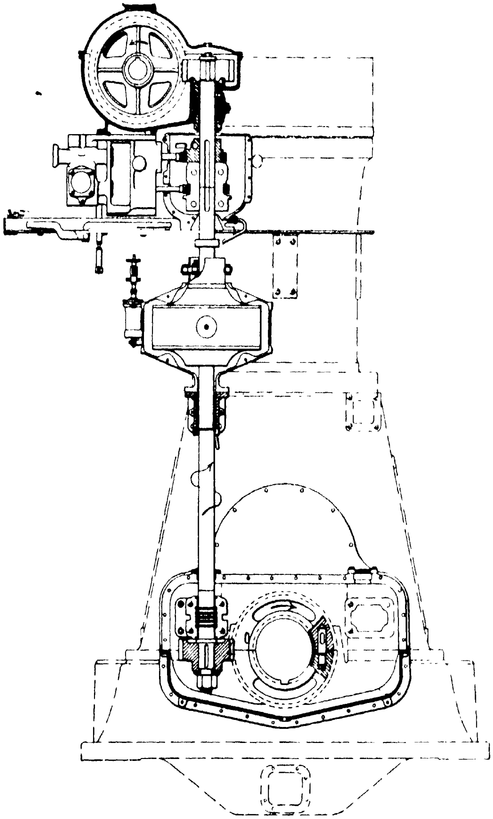


FIG. 254.—Busch-Sulzer governor drive.

An overspeed governor is also placed on the Busch-Sulzer engine. This is of the ordinary centrifugal weight design. If the engine overspeeds, the weight strikes a lever which releases a catch on a relief valve. This allows the oil to by-pass back to the supply.

Snow Diesel Governor.—The Snow engine is equipped with the Jahns governor with the regulator as appears in Fig. 19. On some of the engines, where a great speed range is required, the class D Jahns governor is used. This governor is identical with the class C governor with the exception of the regulator. The regulator spring is located in the governor stand and surrounds the spindle, rotating with the latter. The lower end of the spring rests on a cross-key passing through a rod inside the hollow governor spindle. This rod is fastened to the roll sleeve inside the governor. The upper end is in contact with a ball-bearing collar having two ears passing through slots in the governor pedestal. The pedestal is threaded and has a hand-wheel holding the spring collar ears. Turning the wheel gives the necessary spring compression for any speed within 50 per cent of normal. With this governor the ball-races should be cleaned occasionally to remove dirt and gum.

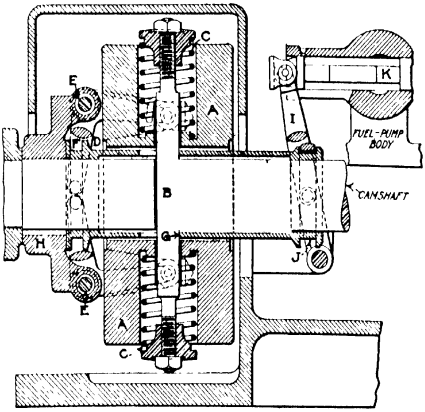


Fig. 255.—McEwen governor.

National Transit Diesel Governor.—The National Transit Co. provide their engine with the Jahns type C governor, Fig. 243, although the class D is used to meet wide speed-range requirements. Figure 244 shows the class D governor connected to the fuel pump.

McEwen Diesel Governor.—The governor of the McEwen Diesel is of the centrifugal weight type. The governor, Fig. 255, consists of two weights *A* that are held together by a spring-loaded rod *B* which passes through the engine layshaft. Two sets of bell-cranks connect the governor weights with the governor sleeve *D* and are fulcrumed on the spider *H*. The sleeve *D* extends along the shaft between the two weights and has a yoke and lever connected to the fuel pump wedge. Any change of engine speed causes the weights to take new positions; this shifts the sleeve along the shaft, causing the fuel pump wedge to shorten or lengthen the pump plunger stroke.

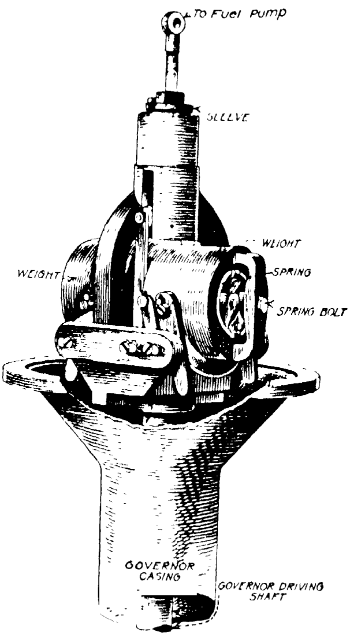


FIG. 256.—Allis-Chalmers governor.

Adjustments.—With this type of governor about the only attention required is the cleaning of all the parts. If the pins and rollers are allowed to become foul with grit and grime, the wear will result in considerable lost motion. This causes the engine to be erratic in speed regulation.

Allis-Chalmers Diesel Governor.—The governor of the Allis-Chalmers Diesel appears in Fig. 256. The governor spindle carries a yoke-shaped frame which revolves with the spindle. To the frame is hung two weights which are held together by two spring bolts and one spring. The spring is in tension, rather than compression as is the usual governor practice. At an increase in the engine speed, which is communicated to the weights through the layshaft and the spindle, the weights move outward. The motion is translated to the regulator connection by the yoke levers. The regulator connection raises the shifts the pump lever as described in Chapter XIV.

This governor has the advantage that lies in the employment of but one weight spring. It is practically impossible to secure two springs possessing the same characteristics. Where two springs are used the outward travel of the two weights will not be equal, throwing a side thrust on the spindle.

Adjustments.—Speed regulation is secured by the spring bolt, while shifting the weight on the outside lever will give a small speed change.

A disk step bearing is fitted to the base of the governor spindle. This consists of two steel disks and one bronze disk. The bronze disk wears from the weight of the governor and the thrust of the helical gear. As soon as the gear emits a grinding sound, the operator should examine the step bearing since, when it wears, it causes the gears to mesh improperly.

Standard Diesel Governor.—The Rites inertia governor Fig. 257 is used on this engine. The governor eccentric gives a variable throw to the fuel pump plunger. The governor is fitted with a dashpot to eliminate the supersensitiveness that is inherent in this form of shaft governor. The governor has the advantage of being extremely simple and devoid of wear with the exception of the governor pin bushing. This must be well lubricated. A change of speed can be secured by altering the spring tension. The regulation can be varied by shifting the location of the spring bolt in the governor slot—moving the bolt in toward the center makes the governor more sensitive.

McIntosh & Seymour Diesel Governor.—This governor, which appears in Fig. 258, is mounted on the vertical governor shaft and is enclosed in a case as shown. The governor consists of a spider *A* which is keyed to the vertical shaft *B* and carries two pins to which are fastened the two weight arms *C*. As the shaft rotates, the weights tend to swing outward, being restrained by the tension of the two springs. One end of the spring is pinned to the weight arm while the other end is held by the spring collar *D* and the bolt to the regulator lever *G*. The lower end of this lever rests on the flat collar *H*, which also rotates with the shaft. The thrust of the collar *H*, due to the reaction of the lever *G*, is absorbed by the spring *I*. A regulator shaft and wheel *J* is provided, a movement of which alters the position of the lever *K*. It is apparent from the drawing that, if the wheel *J* is turned to lift the bearing end of the lever *K*, the thrust of the spring, through the ball-bearing collar, will be less; consequently the collar *H* is raised, throwing the upper end of the lever *G* outward. This increases the tension on the governor springs. Through the control of the weight arms on the fuel pump the engine speed will increase. If this lever *K* moves downward placing an additional thrust on the spring, the latter will be compressed a further amount. This allows *H* to move downward and, in turn, lessens the governor-spring tension, causing the engine to decrease in speed.

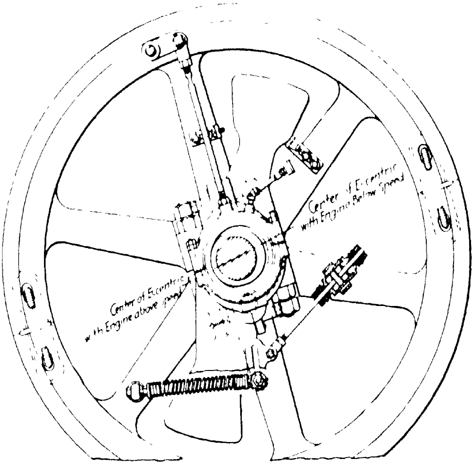

Center of Eccentric with Engine Below Speed
Center of Eccentric with Engine above speed

FIG. 257.—Standard governor

The eccentric *F* is keyed to the shaft *B* and carries a second eccentric *E* which is linked to the governor weights. This eccentric *E* drives the fuel pump plunger through the eccentric strap shown. The eccentric *F*, as it revolves, carries the eccentric *E*, which is held by the equilibrium of the weights and springs,

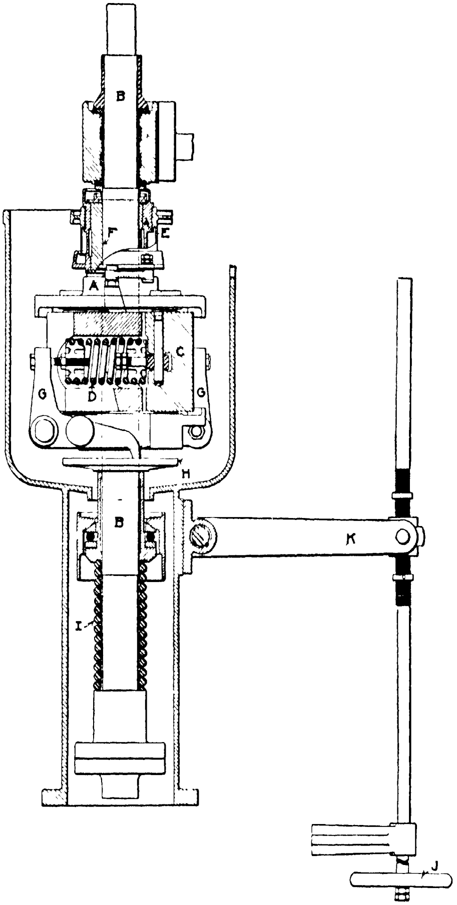


FIG. 258.—McIntosh & Seymour governor.

with it; this causes the pump plunger to reciprocate, thus delivering a fuel charge to the engine cylinder.

Figure 258*A* shows the relative position of the two eccentrics when at rest. Both the left- and right-hand engines have the governor shaft turning counter-clockwise, as viewed from the top. As the engine speeds up, the weight arms pull the eccentric *E* clockwise in relation to the eccentric *F*. This reduces the combined eccentricity of the two eccentrics, which action decreases the pump plunger stroke.

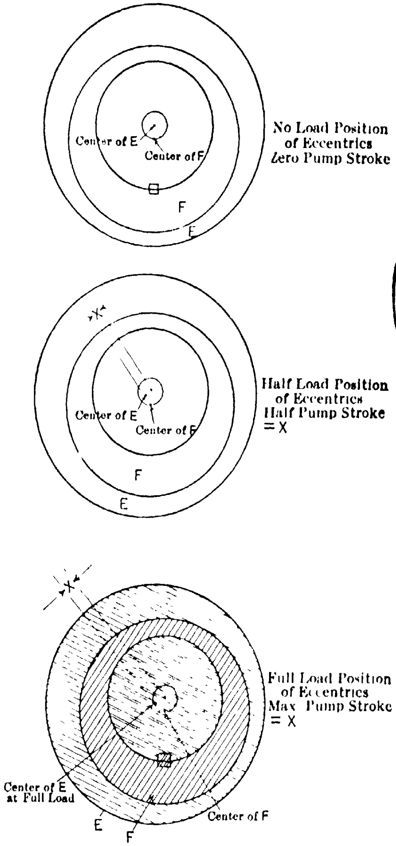

Center of E
Center of F
F
E
No Load Position of Eccentrics Zero Pump Stroke
X
Center of E
Center of F
F
E
Half Load Position of Eccentrics Half Pump Stroke = X
X
Full Load Position of Eccentrics Max Pump Stroke = X
Center of E at Full Load
E
F
Center of F

FIG. 258*A*.—McIntosh & Seymour Type A eccentric movements.

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It will be observed that this acts directly on the pump governor; this places a considerable reaction on the governor and the weights are heavier than is required on a spilling system governing device. The double eccentric design assists materially in reducing the effort required to produce a change in the pump's stroke at a change in speed. The regulation is as close as occurs with other Diesels.

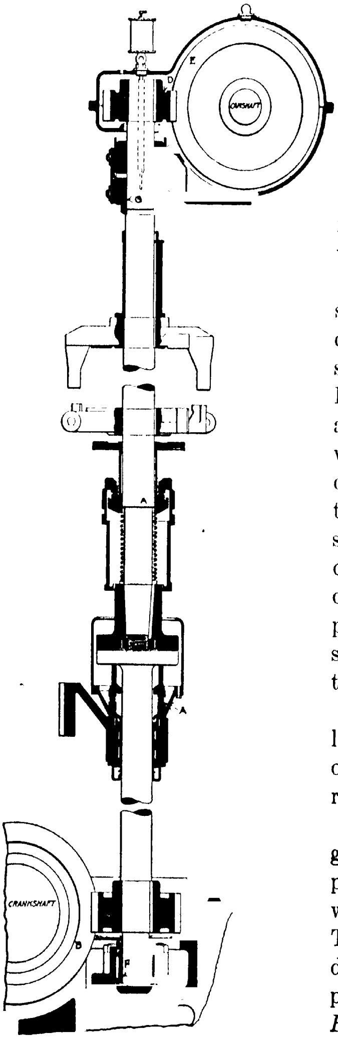


Fig. 259.—Governor shaft drive, McIntosh & Seymour Diesel.

Adjustments.—Since two springs are employed, it becomes necessary to adjust the springs to give equal effect. If the weight-arm positions are not identical, the governor will be out of balance, producing erratic speed regulation. To equalize the two spring tensions, the spring collar *D* can be screwed into one of the springs a fractional part of a turn, thereby increasing the tension on this particular spring.

The eccentrics must be well lubricated to reduce the wear occasioned by the pump reaction.

The weight of the entire governor assembly is supported by the ball bearing *A*, which is shown in Fig. 259. This bearing also receives the downward thrust of the upper set of helical gears *D* and *E*. Precaution must be taken to replace any broken balls, and all wear of the ball-race must be taken up by shimming under the case which is supported by the bracket. If the bearing lowers, the upper gear train will not mesh properly. This produces severe cutting of the gear teeth and increases the side pressure on the upper bearing *G*. If this bearing wears, the shaft is thrown out of plumb, placing a side pressure on the lower bearing *F*. Since this bearing *F* is rather inaccessible, the average operator neglects to inspect it for wear.

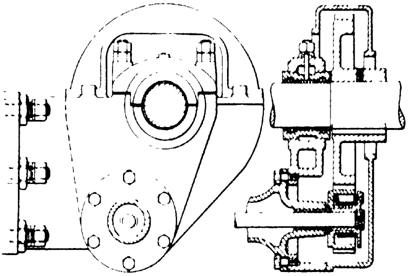


Fig. 260.—Fulton governor drive

Nordberg Governor.—The governor of the Nordberg two-stroke-cycle engines is shown in Fig. 249. This governor, of the Jahns type, is driven by bevel gears from the camshaft. The governor sleeve is connected by a link to a bevel or mitre gear and by moving this gear relative to a driving mitre gear the position of the mitre gear on the fuel by-pass shaft is altered. This causes the by-pass valve to close early or late as the load requires. Speed variation is obtained by moving a weight on a bar connected to the governor sleeve.

Fulton Governor.—The Fulton Diesel uses a Jahns governor of the type previously shown in Fig. 252. The drive, however, differs from that on other engines in that the governor is not mounted directly on the vertical shaft which meshes with the camshaft. Instead, a flexible gear, Fig. 260, keyed on the vertical shaft drives a pinion on the governor shaft. The effect of this arrangement is to prevent the engine pulsations from being transmitted to the governor. The assembly is shown in Fig. 261.

Governor Shaft Gears.—Practically all the Diesels have the governor either mounted on a shaft geared to the engine shaft or are driven through such a shaft. On the vertical engine the governor is customarily mounted on the vertical shaft connecting the camshaft to the engine crankshaft. This involves the use of a set of gears at the top and bottom of this vertical shaft; ordinarily helical gears are employed since the gear diameters can be kept within reasonable limits. With all helical gears there is a thrust in the direction of the driven shaft.

Gear Lubrication.—The lubrication of the governor-shaft and layshaft gears is of vital importance. The majority of engine builders have adopted some form of stream lubrication. If this is to be successful in keeping down gear-teeth cutting, the oil must be fed in a heavy stream—it is useless to deposit the oil on the teeth a drop at a time. Much of the oil never reaches the pressure line between the two contact teeth as the teeth squeeze the oil film out the sides of the gears. When the gears show a tendency to cut, which is due to poor lubrication rather than to play in the shaft bearing, the sole remedy is the adoption of an oil having a heavier body. This oil will not flow off the teeth as readily as does the standard engine oil.

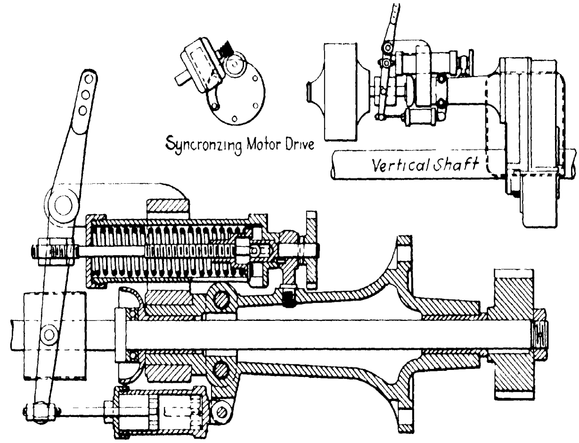


Fig. 261.—Fulton governor.

CHAPTER XVI

AIR-COMPRESSION SYSTEMS

TYPES. ADJUSTMENTS

Air-compression Systems.—The air that is employed in injecting or blowing the fuel charge into the engine cylinder must necessarily be at a high pressure. Since the force that produces the flow of air into the cylinder against the resistance of the fuel valve atomizing disks is the difference in the injection air pressure and the engine compression pressure, it is evident that the air pressure must be above the compression pressure of approximately 500 lb. To insure a thorough nebulization of the oil and a rapid rate of injection, the air pressure must exceed the value by several hundred pounds. The design of fuel atomizer governs the required air pressure; this pressure will range from 1,050 to 750 lb. per square inch at full load.

The production of this extremely high air pressure becomes one of the serious problems of Diesel-engine construction and operation. The ordinary commercial air compressor is of too light a design to guarantee continuous service; consequently all Diesels are equipped with compressors that have been developed especially for this service. While a few of the first Diesels used a single-stage compressor, all modern engines have either a two-, three-, or four-stage compressor. The air temperature due to a final pressure of 900 lb. is high enough to ignite the lubricating oil if any is trapped in the pipe line. To overcome this as well as the objection of having a large diameter compressor cylinder exposed to this high pressure, the compression is divided into stages with intercoolers placed between these stages. This design places but a small pressure on the low-pressure cylinder and allows the temperature of the air, after leaving each stage, to be reduced to approximately the temperature of the outside air. The high-pressure discharge temperature is then around 250 deg. F. rather than 1,500 deg. or more which would exist without inter- or aftercoolers. Some engineers consider that this between-stage cooling reduces the power consumption of the air compressor. Considering the same volume of air compressed with and without intercooling, there is a saving from intercooling. However, since without intercooling either the discharge pressure or discharge volume will be greater than with cooling, a smaller volume of air could be compressed under noncooling conditions to perform the same degree of atomization. The practical advantages of intercooling are the lessened danger of cylinder fracture, the freedom from explosions of lubricating oil that might collect in the discharge piping, and the better service obtained from the compressor valves while working under a fairly small temperature range.

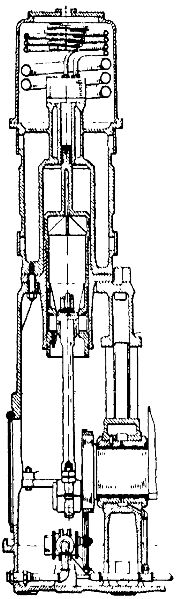


FIG. 262.—Three-stage compressor.

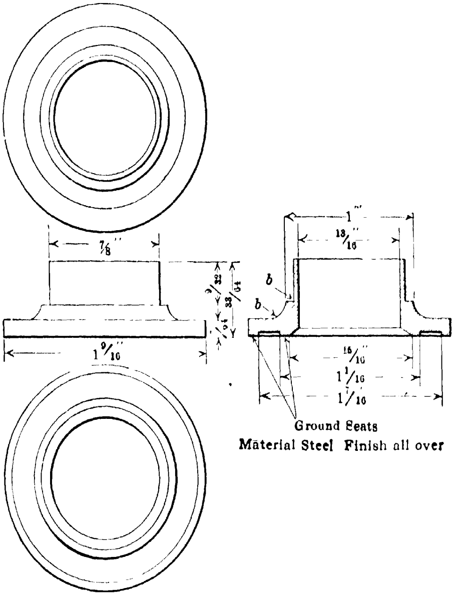
Compressor Stages.—In engines ranging over 150-hp. capacity it is customary to employ a three-stage compressor. With the smaller units a two-stage compressor is more generally used. The volume of air required for these smaller-powered engines is such that the cooling effect of two stages and one intercooler is ample to remove the objectionable temperature conditions. The three-stage compressor customarily has the cylinders placed insteeple or escalon, wherein the high-pressure cylinder is mounted above the low-pressure cylinder while the intermediate compression space is between the bottom of the low-pressure cylinder and the top of the low-pressure piston, the latter being turned with a barrel smaller than the head, as is outlined in Fig. 262.

Independent Air Compressors.—The first Diesels manufactured in the United States were equipped with belt-driven three-stage straight-line air compressors. The difficulties experienced with the breaking and slipping of belts resulted in the adoption of compressors directly connected to the engine crankshaft. The independent compressor has certain features that are of advantage in Diesel practice. If two compressor units are installed with independent motor drive, either belted or geared, there is no danger of plant shut-down owing to compressor trouble. Since at least 50 per cent of the breaks in Diesel operation continuity are attributable to trouble with the air compressor valve, it must be conceded that any system that will enable the engine to continue functioning in the face of a broker air-compressor valve ought to be welcomed. In a number of plants where a modern Diesel with direct-connected compressor is installed by the side of an old Diesel with an independent motor-driven compressor, it is quite common to see the engineer start up the independent compressor to relieve the engine compressor because of a faulty valve. Without the independent compressor such a plant would be helpless in case of accident, if, as is usual, the plant load is so large that it cannot be handled with one engine disabled. For these reasons, each plant should possess one motor-driven compressor of a capacity sufficient to handle the largest engine installed in the plant even though the engines have built-in compressors.

The large marine Diesels are usually supplied by an independent engine-driven compressor. This assures an air supply for maneuvering.

Built-in Compressors.--As has been heretofore mentioned, the modern Diesels have the air compressor built onto the engine frame, being driven by a crank on the end of the crankshaft. As previously outlined, this has the objection of entailing an engine shut-down in case of compressor failure. On the other hand, the direct-connected unit simplifies the plant installation and brings this important auxiliary under the eyes of the engine operator. This insures a better lubrication treatment and places the compressor in a light, dustless location. Moreover, this design entails the employment of a minimum amount of pipe. Engines having the closed-nozzle fuel-injection valve are ordinarily provided with air bottles in which is stored an excess supply of air. These engines can be satisfactorily charged from a separate compressor unit. The engines employing the open-nozzle valve seldom have air bottles; the air enters the atomizer direct from the compressor. With these engines, before an independent compressor can be used, some manner of air bottle must be added to the equipment since the independent compressor never runs at engine speed.

Beyond these, there are no other advantages in the direct-connected compressor for stationary engines. With engines of variable speed, such as marine engines, the separate compressor is essential if maximum economy is to be secured. On slow speed the amount of air needed at the injection valve is the same as at full speed, since the actual time during which the fuel valve is open is constant. If a built-in compressor charged the engine, at high speed an excess amount of air would be compressed and ultimately lost through the relief valve. Since the direct-connected compressor operates at engine speed, the rate of compressor output is greater than the rate of demand.



Ground Seats
Material Steel Finish all over

FIG. 263.—Busch-Sulzer air-compressor valve.

Busch-Sulzer Diesel Air Compressor.—A three-stage air compressor, Fig. 262 is incorporated in the design of the Busch-Sulzer Class B engine. The compressor cylinder assembly is bolted to the engine frame, giving the appearance of a fifth cylinder. The low-pressure cylinder casting is bolted to the frame with the high-pressure cylinder above it while the intermediate cylinder is formed by the stepped piston, being between the piston barrel and the low-pressure cylinder walls. All the valves are similar to the high-pressure suction valve shown in Fig. 263. These valves are of the flat-seated poppet type and are made of alloy steel. The operator of this engine should keep on hand a complete set of spare valves since they, as do most valves, frequently break at the point *b*, Fig. 263. There is serious question as to the underlying cause of this valve fracture; it is probably due to too great a valve lift, resulting in hammer blows as the valve falls back on its seat.

With this air compressor is furnished three air bottles—two for storage and one for the starting air. The air is cooled both between stages and after passing the high-pressure discharge valve. The cooling is effected by copper coils placed in the top of the compressor jacket and surrounding the high-pressure cylinder.

Air Pressure Regulation.—To give a variable air-discharge pressure dependent on the engine load, the suction of the low-pressure cylinder has a damper under control of the engine governor. On low loads the movement of the governor sleeve partially closes the suction opening. This results in a low air-discharge pressure. To the discharge of the low-pressure cylinder in large units is connected the servomotor which controls the fuel valve opening and closing points. As the air suction is throttled, lowering the low-pressure cylinder terminal pressure, the servomotor piston moves under the influence of a spring which overcomes the lessened air pressure on the opposite side of the servomotor piston making the fuel valve open late and close early.

Type C Compressor.—The air compressor, Fig. 264, which provides compressed air for fuel injection and starting, is mounted vertically on the crankcase at the forward end of the engine in line with the working cylinders. It is also driven from an extension to the main crankshaft. The compressor is three-stage, water-jacketed and provided with intercoolers and aftercoolers to keep the air at a low temperature. Each stage of the compressor is protected against excessive pressure by a safety valve. A regulating device is provided to adjust the injection air pressure to suit operating conditions. This is accomplished by varying the opening of the suction damper valve on the low-pressure stage.

Scavenging Pump.—For providing low-pressure scavenging and charging air for the working cylinders, a scavenging pump, Fig 265, is mounted vertically on the crankcase at the opposite end from the flywheel, next to the forward working cylinder and in line with it. The pump is directly driven from an extension to the main crankshaft and is provided with a crosshead and guide similar to those of the working cylinders. Both suction and discharge valves, Fig. 266, are of the simple automatic shutter type mounted in cages. These valves are identical in size and design and are interchangeable. The discharge side has a valve chest with connections to the scavenging-air receiver, which in turn provides the connection to the working cylinders. This receiver extends along the front of the engine and is bolted to faces on the cylinder jackets. This provides a support for the valve gears, camshaft bearings and casings.

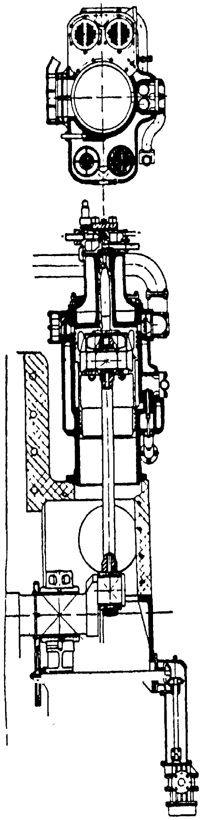


Fig. 264.—Busch-Sulzer Type C compressor.

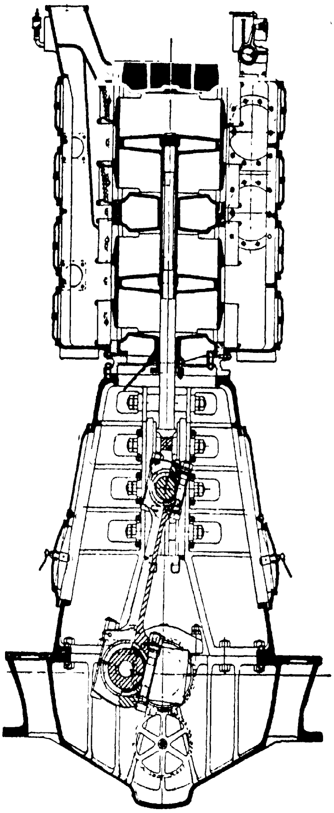


Fig. 265.—Scavenging-air pump, Busch-Sulzer Type C engine.

McIntosh & Seymour Marine-Diesel Air Compressor.—Figure 267 outlines the compressor of the McIntosh & Seymour marine Diesel. The compressor is three-stage and is placed at the front end of the engine, being driven by an overhung crank on the engine shaft. The low-pressure and intermediate cylinders are cast together as shown in Fig. 267, while the high-pressure cylinder is in one piece with the low-pressure cylinder head.

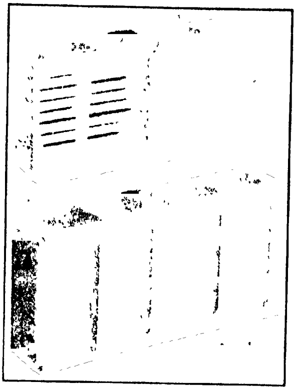


FIG. 266.—Valves of Type C scavenging-air pump.

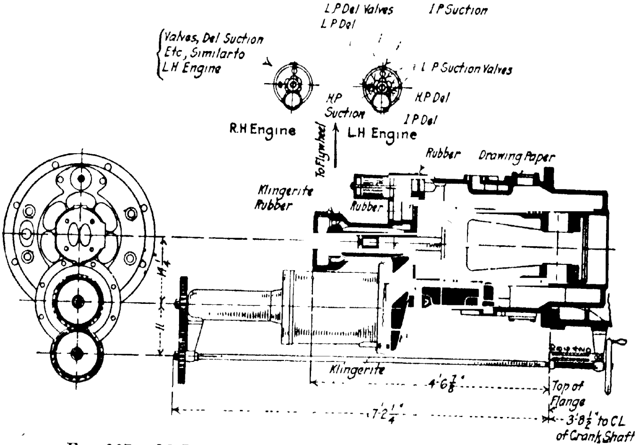


FIG. 267.—McIntosh & Seymour marine Diesel air compressor.

The low-pressure suction and discharge valves are all in the low-pressure cylinder head; the intermediate-pressure valves are arranged around the cylinder barrel and the high-pressure valves are in the high-pressure cylinder head. The air pressure is altered by the control rod acting through the gears at the top of the regulator. By the proper displacement of this piston the low-pressure cylinder volume is altered to give any desired high-pressure discharge pressure. The indicator scale on the handwheel shaft is marked with the same load points as is the fuel-control lever. The operator, then, on throwing the fuel-injection control to full load moves the air indicator to the corresponding point. This gives maximum air pressure for a maximum fuel charge.

The compressor valves closely follow the design appearing Fig. 263. The material is a nickel-steel alloy heat-treated and toughened. A valve lift of 0.03 in. is about the most advisable value to use.

McIntosh & Seymour Stationary-Diesel Air Compressor.—The stationary engine's compressor is practically the same as the compressor of the marine engine, though the regulator is not included on the Type *A* and *B* engines.

The chief points of attention are the valves and the connecting-rod boxes. The former frequently break at the junction of the "brim and crown." This is usually attributed to an excessive lift of the valve and can only be avoided by valve replacement when the lift becomes more than 0.05 in. To avoid air leakage the valve seats must be well-nigh perfect. To regrind a valve only a small amount of powdered glass and vaseline should be used. The slightest scratch on the seat surface will destroy the valve's usefulness.

The connecting-rod brasses require attention more than do the main engine rods. The current of air passing through the compressor frame deposits dirt and grit on the bearings that soon reveal their presence by overheating.

The control of the air pressure is attained by the regulation of the valve leading to the starting tank; opening this allows part of the air compressed to enter this tank, lowering the pressure in the line to the fuel pump. Two running air tanks and one starting tank are supplied for each engine.

The large crosshead type McIntosh & Seymour Diesels have an air compressor similar to that of the marine Diesel. The method of regulation by means of a movable piston in an auxiliary receiver or cylinder is followed.

Snow Diesel Air Compressor.—The Snow horizontal Diesel is equipped with a three-stage compressor, the piston being of the stepped type, although the smaller units are fitted with two-stage compressors. The three-stage unit cross-section appears in Fig. 268. The valves are of the flat feather type, which are of light steel strips restrained at the ends and free to lift from the seat to allow the air to pass. This valve design permits the valve to seat fairly gently without danger of fracture due to hammering; the fractures that do appear are caused by "fatigue," resulting from continual bending of the steel fibres.

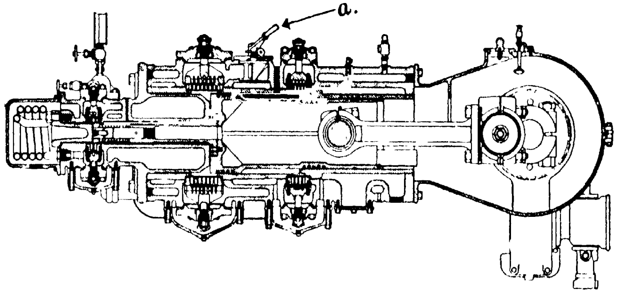


FIG. 268. Snow three-stage air compressor.

The compressor is arranged with intercoolers between the stages. These are formed in the cylinder jackets. The high-pressure discharge passes through a cooling coil placed in the high-pressure cylinder cover.

Pressure control is obtained by altering the low-pressure suction area through the hand lever *a*.

Snow Vertical Diesel Air Compressor.—This three-stage compressor is quite like the compressor in Fig. 269, although the coolers are in the jacket of the compressor.

Worthington Marine Diesel Compressor.—This marine Diesel has a three-stage compressor, Fig. 269, driven by a crank coupled to the end of the engine crankshaft. A special feature is the outer bearing for the support of the compressor crank. In this way the bending of the shaft is eliminated.

McEwen Diesel Air Compressor.—On the small- and medium-powered McEwen Diesel engines a two-stage compressor is

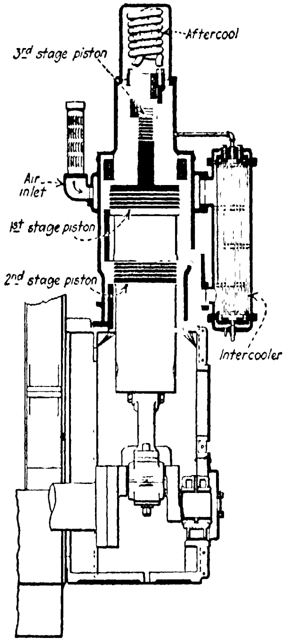


FIG. 269.—Worthington air compressor.

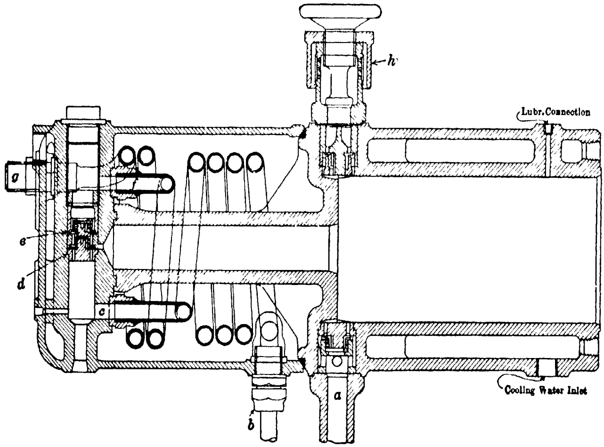


FIG. 270.—McEwen air compressor.

mounted on the side of the engine frame, Fig. 270. This stepped piston is driven by a crank on the end of the engine shaft. The valves are of the standard "plug hat" form. The two cylinders are water-jacketed, and the intercooler and aftercooler coils, of copper tubing, are placed in the high-pressure cylinder jacket. The air, after leaving the low-pressure discharge at *a*, passes through an oil separator before entering the intercooler coils at *b*. After being cooled, the air issues from the coil at *c* and enters the high-pressure cylinder through the high-pressure suction valve *d*. After being compressed, it is forced out the discharge valve at *e* into the aftercooler coils at *f*. It leaves these coils at *g* and flows to the fuel valve.

Pressure control is achieved by regulating the suction opening at *h*. The valve lifts should be between 0.03 and 0.04 in.

No air tank is interposed between the compressor and the fuel valve; consequently the air charge for both compression strokes of the air compressor enters the fuel nozzle on the power stroke of the engine. The air line is of some volume; therefore the drop in the discharge air-line pressure at the point of injection valve opening is but slight.

National Transit Diesel Air Compressor.—The National Transit Diesel has a three-stage compressor bolted to the engine frame, the piston being stepped, and driven by a crank on the end of the engine shaft. The two cylinders and the valve-cavity covers are water-cooled; an interreceiver between the two cylinders serves as an intercooler as well. Regulation is obtained by hand control of the suction opening.

Standard Fuel Oil Diesel Air Compressor.—A two-stage compressor is driven by a crank off the engine crankshaft. While the compressor is only two-stage, it has a three-stage effect. The air is compressed in the scavenging cylinder of the engine to from 7- to 11-lb. gage. It then passes into a receiver from where it is blown into the engine-working cylinder during the exhaust portion of the cycle. The suction of the low-pressure air-compressor cylinder is connected to this scavenging air receiver; consequently the low-pressure cylinder is in effect an intermediate cylinder.

The discharge from the low-pressure cylinder is piped to the starting air tanks. A connection on this line leads to the pressure control, Fig. 271. The air enters the control body at *a*. The plunger is under the control of the engine governor, altering its stroke according to the load carried. As the plunger moves to the right, the port *b* is uncovered, and the air flows through the port *b* back to the suction of the high-pressure cylinder of the air compressor where it is further compressed to about 900 lb. From the high-pressure discharge the air is forced into the fuel valve.

It is apparent that, with the larger volume to be compressed, the final pressure at the fuel valve is high, as is desirable, at full load. On lower loads the plunger uncovers less of *b*, allowing a corresponding smaller amount of air to enter the high-pressure cylinder, reducing the injection pressure. This injection control is very close in regulation; a hand adjustment allows any desired pressure to be secured at any set governor position.

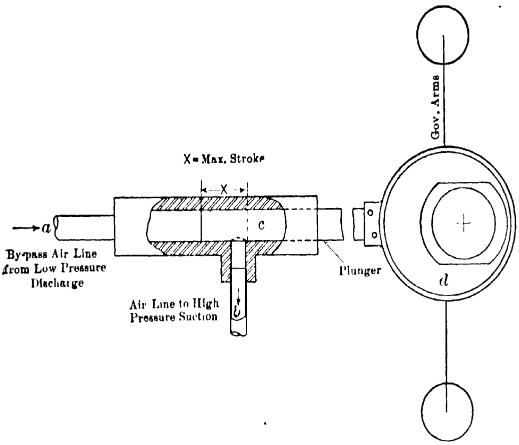

Gov. Arms
X = Max. Stroke
X
a
c
By-pass Air Line from Low Pressure Discharge
Plunger
d
Air Line to High Pressure Suction
b

Fig. 271.—Standard horizontal Diesel injection air pressure control.

To prevent an oversupply of air on low loads the first-stage suction has a choke control.

Standard Vertical Diesel Compressor.—The high-pressure air required for injecting the fuel into the combustion space is furnished by a two-stage compressor, Fig. 272, which is mounted on one end of the box frame and is driven directly from the crankshaft.

A stepped piston is used to obtain the necessary displacements for the different stages.

Both the low stage suction and discharge valve are located in a single pocket at the side of the cylinder. This pocket is so positioned as to make it impossible for a valve to get into the cylinder due to either a broken valve spring or stem and yet the clearance volume is less than 2 per cent of the cylinder displacement.

The valve cage for the high stage valves is located in the upper end of the cylinder bore and with its cover forms the cylinder head. This cage is also arranged so that there is no possibility of a valve getting into the cylinder and wrecking the machine.

After being compressed in the first cylinder and before going to the second the air is cooled in an intercooler consisting of coils of copper tubing and enclosed in the water-jacket space of the compressor cylinder through which all the cooling water passes before going to the main cylinders. An aftercooler of the same design is used to cool the high pressure discharge air.

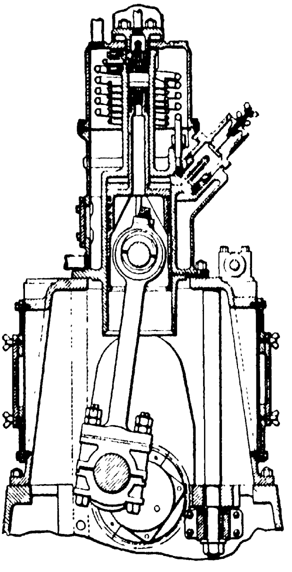


Fig 272.—Standard vertical air compressor.

Scavenging Air Pump.—The Standard engine has a stepped main piston, and the scavenging air is compressed in the space between the cylinder and the lower enlarged piston end, see Fig. 70*A*.

The valves are of the plate type and are placed outside the cylinder casting in the air manifold. Access to the valves is secured by removing a light cover plate.

Allis-Chalmers Diesel Air Compressor.—The 14- and 16-in. stroke Allis-Chalmers Diesels employ a two-stage air compressor, while the 18-in. stroke units have three-stage compressors. The designs of the two- and three-stage compressors follow the same general lines. These units differ from most Diesel compressors in that the intercoolers are not placed in the high-pressure cylinder head jacket. Each cooling coil is placed in a cavity in the base of the compressor casting, which is in the form of a cast-iron drum. The air from the low-pressure discharge enters an air pot or receiver; a copper pipe is coiled about the pot and is connected to it. Both coil and pot fit into the cooling drum. In Fig. 20 the compressor is shown in position on the engine frame. On the two-stage compressor the air, after passing through the inter-cooler coil and pot, is compressed in the high-pressure cylinder; the air is then discharged into the aftercooler air coils which are placed in the aftercooler water pot. On the three-stage compressors two intercoolers are used. The valve and cage appear in Fig. 273.

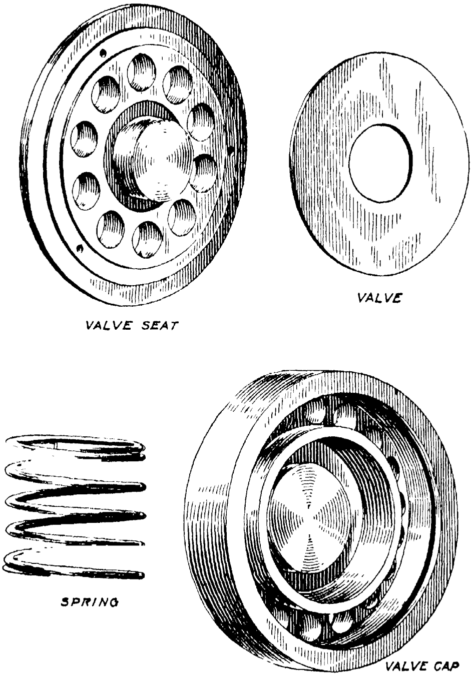


Fig. 273.—Allis-Chalmers Diesel air-compressor valve.

Nordberg Type FD Air Compressor.—The three-stage compressor of the later larger Nordberg two-stroke-cycle Diesels is shown in Fig. 274. This is placed at one end of the engine and is driven by a small crank on the crankshaft. As shown in the illustration, the low- and high-pressure cylinders are above the piston and the intermediate cylinder in a recess below the piston head. For both suction and discharge circular plate valves are used, these being held to their seats by light springs.

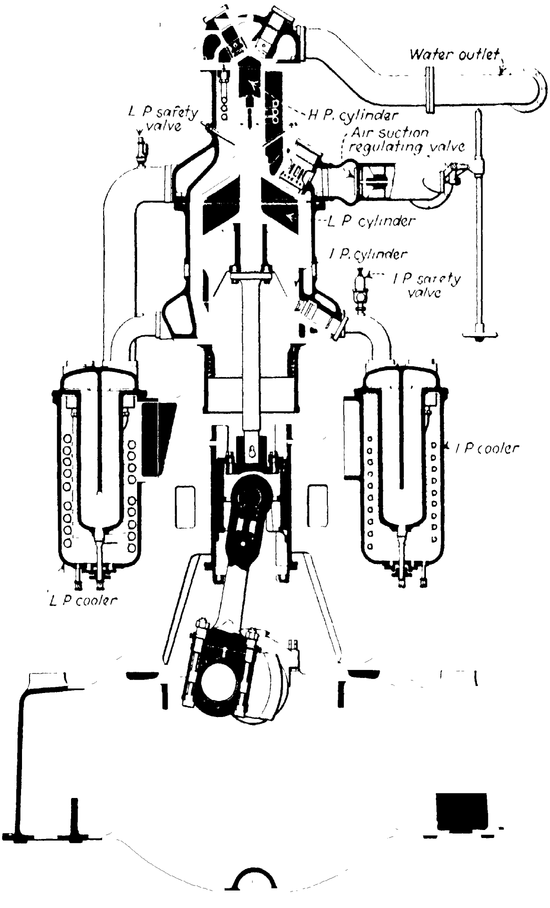


FIG. 271.—Compressor of Nordberg FD engine.

The quantity of air is controlled by means of a hand-regulating valve on the suction. Between each stage the air passes through intercoolers, and the high-pressure discharge is cooled by an aftercooler consisting of a pipe coil in the high-pressure water jacket.

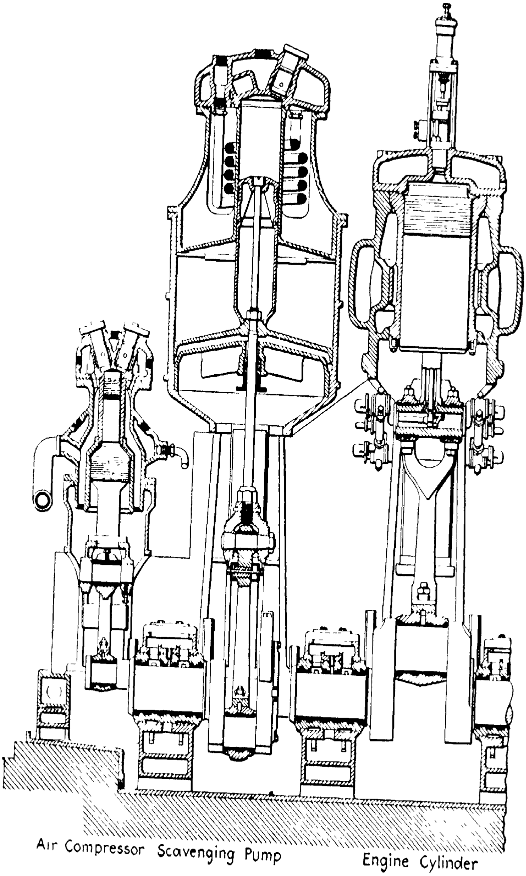


FIG. 275.—Nordberg Type VE compressor and scavenging pump.

The intercoolers are provided with drips which must be opened about once an hour to remove the moisture thrown down by the air in cooling.

Type FD Diesel Scavenging Pump.—The scavenging pump is mounted on the engine bed between the air compressor and working cylinder. This pump is of the vertical double-acting type; the suction and discharge valves are the Nordberg plate type. The discharge pressure ranges from 4 to 6 lb., and the air, upon leaving the compressor, passes through a header along the engine cylinder. This header has a connection to each cylinder head.

Nordberg Type VE Diesel Compressor.—Starting and blast air is furnished by a three-stage compressor, equipped with three air coolers. One of the coolers takes care of the air coming from the low-pressure stage, another of the air coming from the intermediate pressure cylinder and the third is the aftercooler.

As shown in Fig. 275, the low-pressure cylinder is mounted above the scavenging pump and is driven from the same crank as the latter. The intermediate and high-pressure pistons are driven directly from the crankshaft through a crosshead. The valves in the compressor are of the round plate type, held in their seats by light springs. The compressor is built to compress air to 1,000-lb. end pressure to charge the starting-air bottles. The air pressure can be regulated and adjusted to the needs of injecting the fuel oil into the cylinders. The intake air passes through a suction silencer, consisting of sheet-iron plates mounted one on top of the other at the corners, similar to log-house construction, so that the small passages provided subdivide the air currents and therefore practically eliminate the suction noise.

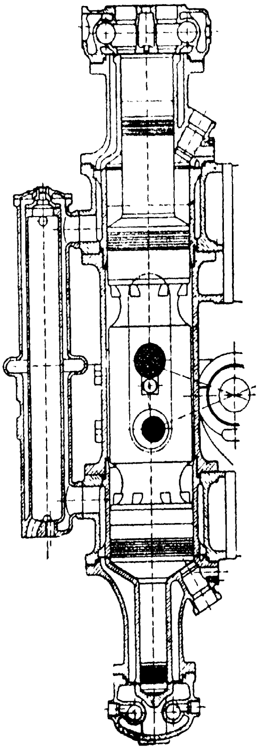


FIG. 276. Type EG compressor.

Nordberg Type EG Air Compressor.—The type EG Diesels, having cylinders of 250 hp. each, have an air compressor similar to that shown in Fig. 275*A*. This is a three-stage machine and is driven by a linkage from the scavenging-pump crosshead.

Fulton Compressor.—The compressor of the Fulton Diesel is shown in Fig. 277. This three-stage air pump is driven off an overhung crank at one end of the engine crankshaft, the crank being a separate forging bolted to the end face of the shaft. The valves are the light plate type and are placed in cages easily removed. The high-pressure discharge valves are of the mushroom type and are set in the cylinder head. The delivery pressure is controlled by a hand-operated suction-choke valve. The intercoolers are double-walled cast-iron tubes with water flowing down the inside tube while the air flows along the passage between the inside tube and the outer tube or casting. These castings are placed in the intercooler compartment. In this way water is in contact both with the outside and interior of the cooling chambers. The aftercooler is a pipe coil in the high-pressure cylinder water jacket.

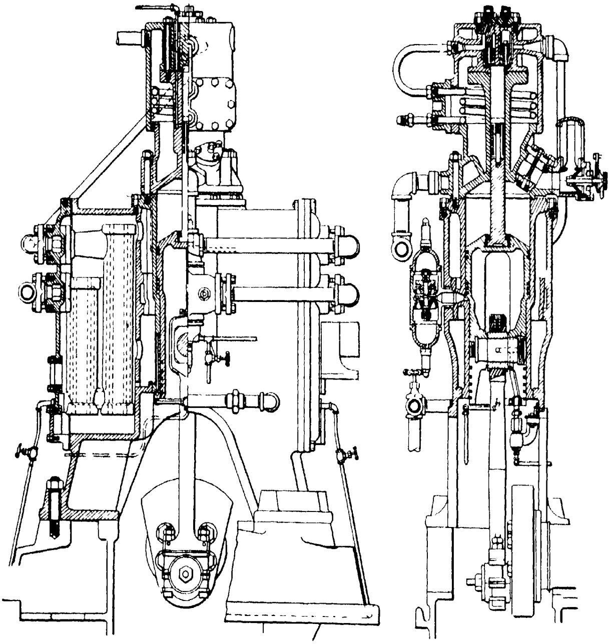


FIG. 277.—Fulton compressor.

New London Compressor.—The air compressor used on the New London Diesel may be seen in Fig. 72: Figure 278 shows exterior view of the same compressor. The compressor is three-stage and a somewhat unusual feature is the crosshead.

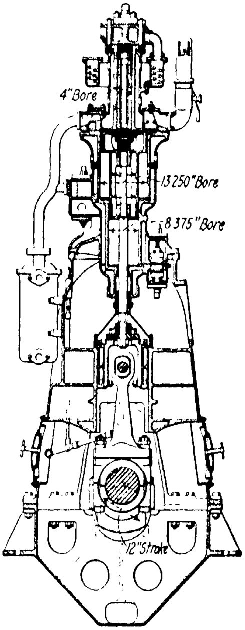


Fig. 278.—New London compressor.

Cramp's B. & W. Compressor.—The compressor of the Cramps marine Diesel is mounted on the end of the frame and is driven by a crank on the engine shaft. A cross-section of the unit is shown in Fig. 278*A*.

Compressor Valves.—The engineer should early recognize the serious part the air compressor and its valves play in the successful operation of the engine. The compressor must function continuously and perfectly. To attain this, the air valves must be kept in first-class condition.

The majority of operating troubles experienced with the valves are due to one of the following causes: Dirty, warped or broken valves; leaky gaskets under the valve cages; carbonized oil on the valve or valve seat. Because of this fact a great deal of ingenious thought has been applied to the improvement of valve design, resulting in a large number of different kinds of valves.

The operator's surest guide to the condition of his compressors is the gage board. The pressure gages for the different stages should be conveniently located at the operating station where they can be easily seen. Any change in pressure ratios as shown by the gages is a sure indication of some changes in conditions within the compressor or in the air system. The immediate result of leaky valves, whatever the cause of leakage, is reduction of capacity, followed by abnormal heating of some part of the compressor. If the capacity reduction proceeds far enough, the engine will become smoky, due to low air pressure and imperfect combustion, and finally stop for want of sufficient air to spray the fuel into the cylinders.

Fulton Compressor.—The compressor of the Fulton Diesel is shown in Fig. 277. This three-stage air pump is driven off an overhung crank at one end of the engine crankshaft, the crank being a separate forging bolted to the end face of the shaft. The valves are the light plate type and are placed in cages easily removed. The high-pressure discharge valves are of the mushroom type and are set in the cylinder head. The delivery pressure is controlled by a hand-operated suction-choke valve. The intercoolers are double-walled cast-iron tubes with water flowing down the inside tube while the air flows along the passage between the inside tube and the outer tube or casting. These castings are placed in the intercooler compartment. In this way water is in contact both with the outside and interior of the cooling chambers. The aftercooler is a pipe coil in the high-pressure cylinder water jacket.

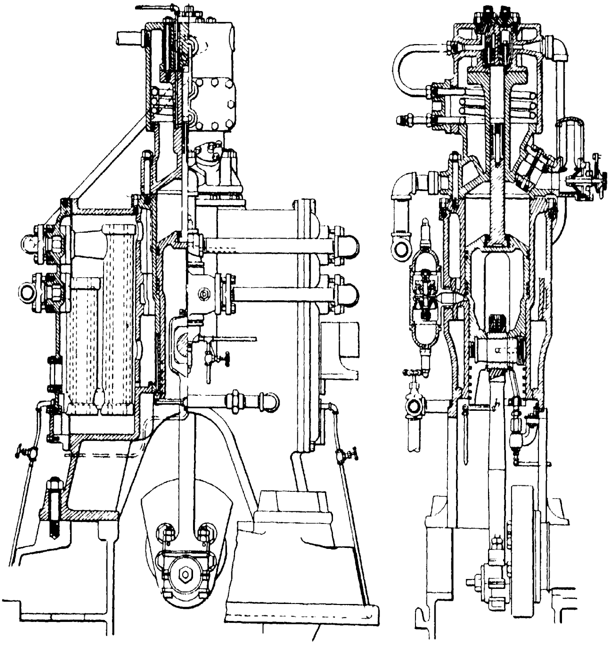


FIG. 277.—Fulton compressor.

possible to keep the engine operating, although the compressor temperature will be higher than usual, especially at the leaky valve cage; but if the leak is large enough, the pressure will rise rapidly and the relief valve on the affected stage will blow. When this happens, the compressor must be stopped until the defective valve can be repaired. In case an auxiliary compressor is available and it is not desirable to stop the engine, the defective compressor may be cut out by closing the suction to the first stage and operations continued, using air from the auxiliary. The rapidity of pressure rise will offer an indication as to the nature of the trouble, a slow rise indicating leakage due to slightly defective valve seats, dirt under valves, warped valves or defective gaskets and a sudden, rapid rise indicating a broken or stuck valve. When the normal operating temperature has been exceeded, it is well to examine the valve springs at the first opportunity, as the heat will often take all the temper out of the springs and render them ineffective. In most compressors the valves are contained in cages and seat on the cages, so that destruction of a valve seat does not involve renewal of a compressor cylinder or head. These cages rest on seats in the cylinder or head casting, and a tight joint is made by means of a gasket. These gaskets frequently leak, due to being burned in service or being carelessly put in. Copper or paper makes the best gaskets for these joints, and in general the thinner the gasket the better it will hold. The most satisfactory joint, however, is made by removing the gasket and using a metal-to-metal joint.

Whenever the engine is stopped, the discharge valves in the high stage can be quickly tested for tightness by leaving the air line between spray bottles and compressor open and opening the indicator cock on the high-stage cylinder. If air blows through, the discharge valves are leaky. If the indicator cock is closed and the second-stage gage shows pressure, the high-stage suction valves also leak.

Regrinding Valves.—In regrinding all types of valves, a very meager amount of powdered glass and vaseline is used. The valve should be rotated very lightly, with no pressure exerted on it and the valve cage. After grinding, all parts must be thoroughly washed with kerosene or gasoline. A defective valve reveals itself in the change in the air pressure.

Air Suction.—If the plant is at all dusty, an air pipe with the free end covered by a fine mesh screen should be run from outside of the building. If this is not possible, the suction connection at the compressor should be screened with a fine-mesh netting, covered with muslin. The dust will collect on the screen, and the engineer need not be informed that he must clean this screen at least once a week.

Air-compressor Lubrication.—This will be more fully discussed in the chapter on Lubrication. The engineer is quite safe in following factory instructions as to the amount of cylinder lubrication he should use. It is not advisable to increase the amount recommended, since any oil that is not destroyed in the cylinder will collect in the air piping. This oil deposit will quite likely explode, especially if the air temperature is not maintained below 150 deg. As a safeguard, the discharge line should be equipped with an oil separator and relief valve. The latter is absolutely necessary. A precaution that can well be observed in every plant is the daily purging of the air line and bottles of all collected oil and moisture.

Intercooler Tubes.—When salt water or water with high acid content is used for cooling, it may sometimes happen that the intercooler tubes will become pitted through and air will be discharged into the water space, stopping the flow of water and causing the compressor to overheat. In the case of vertical engines leakage of air into the cooling water will be shown by a large amount of air collecting in the water-jacket space in the cylinder heads. A check on this is obtained by frequently opening the air cocks on the cylinder heads. The cooling of the air should be watched carefully and it should always be borne in mind that the temperature of the discharge air has a direct relation to the temperature of the suction air. A pretty good indication as to general operative condition of the compressor can be obtained from the thermometer in the discharge line from the compressor. If the temperature here rises above normal, conditions are wrong somewhere and an investigation is in order.

Compressor Clearance.—Whenever the compressor bearings have been adjusted, the engine should not be started until the linear clearances of the compressor pistons have been checked. These clearances are sometimes as little as $\frac{1}{32}$ in., and a very slight bearing adjustment will cause a large clearance change.

Injection Air Pressures.—It is usual to carry air pressures from 700 to 1,000 lb. per square inch. Each design of fuel atomizer requires a certain air pressure to inject the oil properly. This pressure will vary with the resistance or "braking" action of the atomizer disks. It follows that no set pressure values can be given which will apply to all engines. However, the curve in Fig. 229 gives the relation between engine load and injection pressure that is correct for practically all closed-nozzle engines. For open-nozzle units the pressure will run some 50 to 100 lb. lower than is given in the table.

On low loads the amount of fuel deposited in the spray valve is small. Since in most engines the time during which the spray valve is open is constant regardless of the load, there will be a large amount of air blowing into the cylinder after the oil has all been injected. This air will chill the burning oil and decrease the power. In addition, since the work of compressing air is not at 100 per cent efficiency there is a loss in engine efficiency. By lowering the air pressure on light loads the air loss is less. Besides a small oil charge does not offer as great a resistance to flow as does a large charge—a high air pressure is not needed.

Compressor Wear.—The compressor cylinder and piston does not experience as much wear as the main engine pistons. Due to the air loss with only a nominal clearance, a given wear has a greater ill effect in the case of the compressor. The wear is often found to be in a direction parallel to the shaft. This is caused, undoubtedly, by sideplay of the engine shaft. The Diesel Engine Users' Association reports the wear in one 4-in. compressor after 6,000 hr. of service to be as follows:

Position	Wear parallel to shaft, in.	Wear at right angles to shaft, in.
End	0.002	0.006
1½ in.	0.0025	0.004
12 in.	0.0035	0.002

Capacity of Compressor.—The compressor capacity per engine horsepower varies over a wide range, depending upon the builder, etc.

A formula which gives a safe value is

Air Suction.—If the plant is at all dusty, an air pipe with the free end covered by a fine mesh screen should be run from outside of the building. If this is not possible, the suction connection at the compressor should be screened with a fine-mesh netting, covered with muslin. The dust will collect on the screen, and the engineer need not be informed that he must clean this screen at least once a week.

Air-compressor Lubrication.—This will be more fully discussed in the chapter on Lubrication. The engineer is quite safe in following factory instructions as to the amount of cylinder lubrication he should use. It is not advisable to increase the amount recommended, since any oil that is not destroyed in the cylinder will collect in the air piping. This oil deposit will quite likely explode, especially if the air temperature is not maintained below 150 deg. As a safeguard, the discharge line should be equipped with an oil separator and relief valve. The latter is absolutely necessary. A precaution that can well be observed in every plant is the daily purging of the air line and bottles of all collected oil and moisture.

Intercooler Tubes.—When salt water or water with high acid content is used for cooling, it may sometimes happen that the intercooler tubes will become pitted through and air will be discharged into the water space, stopping the flow of water and causing the compressor to overheat. In the case of vertical engines leakage of air into the cooling water will be shown by a large amount of air collecting in the water-jacket space in the cylinder heads. A check on this is obtained by frequently opening the air cocks on the cylinder heads. The cooling of the air should be watched carefully and it should always be borne in mind that the temperature of the discharge air has a direct relation to the temperature of the suction air. A pretty good indication as to general operative condition of the compressor can be obtained from the thermometer in the discharge line from the compressor. If the temperature here rises above normal, conditions are wrong somewhere and an investigation is in order.

Compressor Clearance.—Whenever the compressor bearings have been adjusted, the engine should not be started until the linear clearances of the compressor pistons have been checked. These clearances are sometimes as little as $\frac{1}{32}$ in., and a very slight bearing adjustment will cause a large clearance change.

Stage Cooler	Square Feet of Coil Surface per Cubic Foot of Free Air Compressed per Minute
Low pressure	17
Intermediate	4
High pressure	4

Air Reservoirs.—Modern engines are usually fitted with three air tanks, one of these being kept on the air line while the engine is in operation; the other two are used for starting and as a reserve. The receiver capacity varies greatly with engines of different makes. In all cases there should be at least 10 times the stroke volume of one engine cylinder. The arrangement of the air tanks and their gages, placed close to the engine, is shown in Fig. 279. This view was taken at the Calumet sewer plant, containing four 750-hp. Busch-Sulzer Type C engines.

Fig. 279.—Arrangement of air bottles and gages.

Moisture in Injection Air.—Many operators who have not studied the problem are unable to understand why there should be moisture deposited in the air lines after compression by the air compressor. This action is due to the vapor pressure of moisture in the air. Taking 1 cu. ft. of space which may be assumed to be filled with air at atmospheric pressure, 0 gage (14.7 lb. absolute pressure). If this air is at 60 deg. F., the maximum pressure that can be exerted by the water vapor in this cubic foot is 0.25 lb. absolute. The maximum weight of water vapor that may exist in the cubic foot of space is 0.0008 lb. As the temperature is increased the vapor pressure is raised and the weight of water vapor will also change. The vapor pressure and weight of vapor per cubic foot of air at various temperatures are as follows:

Temperature Deg F.	Vapor Pressure Lb per sq in.	Wt of Water Vapor per cu ft
60	0.25	0.0008
80	0.50	0.0015
100	0.94	0.0028
150	3.70	0.0103
200	11.52	0.0298

The air entering the compressor is by the action of compression reduced in volume. As example, 100 cu. ft. is reduced to a volume, calculated by the equation $P_1V_1 = P_2V_2$, of $14.7 \times 100 = 900\ V_2$ or $V_2 = 16.3$ cu. ft., assuming that the intercoolers and aftercoolers bring the air temperature back to 60 deg F., its original temperature. By the table at 60 deg. the amount of water vapor in a cubic foot is 0.0008 lb. The 100 cu. ft. of suction air had 0.08 lb. of water vapor. Consequently the difference in weight ($0.08 - 16.3 \times 0.0008 = 0.06696$) will settle out as water in the discharge pipe and receivers. It happens that the amount of water vapor in the atmosphere is never the maximum amount (that is, the humidity is under 100 per cent) and so the weight of water thrown down per 100 cu. ft. of air compressed is less than shown above, the value given being the maximum possible amount. The operator will see the advisability of frequently opening the water-bleeder valves on the air receivers. Stoppage of the engine due to failure to remove the water is by no means unheard of. At times when moisture reaches the fuel valves the engine will pound. This is caused by the water delaying the ignition of the fuel after it enters the cylinder. When ignition occurs the crank is past dead-center and the engine pounds.

CHAPTER XVII

COOLING SYSTEMS

TYPES OF SYSTEMS. PUMPS. WATER PURIFICATION

Distribution of Heat Losses.—Various experiments on the subject of Diesel heat losses check very closely as to final results. The consensus of opinion is that the total heat evolved in the combustion of a charge of oil in the engine cylinder is absorbed in doing work and in various losses in the engine at the following percentages:

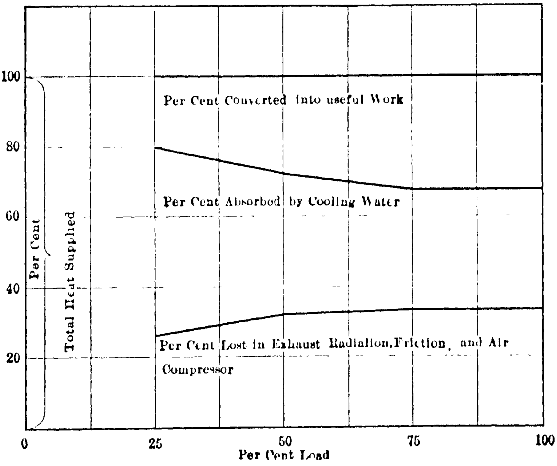


FIG. 279A.—Heat distribution in Diesel engines.

Heat generated in the cylinder	100
Heat converted into work	30
Heat lost in engine friction	6
Heat lost in the exhaust gases	28
Heat absorbed by the cooling water	34
Heat lost by radiation, etc	2

Figure 279*A* covers the losses at various loads. These values are the result of a number of experiments on various engines. If

the work done is 30 per cent of the heat generated, then the engine will consume 8,470 B.t.u. per brake horsepower. The cooling water must be of a quantity sufficient to absorb 34 per cent of the amount or 2,879 B.t.u. in round numbers 3,000 B.t.u. per hour.

Cooling Water Required.—The calculations necessary to employ in determining the amount of cooling water required for a Diesel are quite simple. As an example of the maximum amount that could be used, the intake water temperature at the jacket entrance can be taken as 90 deg. This is at least 10 deg. higher than normal, even with a cooling pond. The engine discharge water temperature can be taken as 140 deg. F., which is 20 deg. below the value most successfully used. Then the rise in temperature will be 50 deg.; consequently each pound of water will absorb 50 B.t.u. The water per hp. hour will be

$$\frac{3,000}{50} = 60 \text{ lb.}$$

This expressed in the form of an equation appears as

$$W = \frac{XH}{100\,(t_1 - t_2)}$$

where

W = weight of water required per hp. hour.
X = percentage of heat absorbed by the water.
H = total heat supplied to the engine.
t_1 = discharge water temperature deg. F.
t_2 = intake water temperature deg. F.

The inlet and discharge temperature given may not check with those observed in any particular installation, nevertheless the temperature range is approximately correct; it is this factor that is important. Table XXV gives the temperatures and quantities of water passing through Diesel engines. This table is the result of a test on three McIntosh & Seymour 500-hp. Diesels installed by the Texas Light and Power Co. at Paris, Texas. Since these engines were developing close to 500 hp. each, this value may well be assumed in computing the water rate per brake horsepower. With this assumption the water per brake horsepower per minute was $1^{25}/_{100}$ lb. or 75 lb. per brake horsepower per hour. This considerably exceeded the value computed above, which can be explained on the grounds that both the exhaust head and air compressor were maintained at a very low temperature. In ordinary operation these engines tested do not carry the exhaust header temperature lower than 130 to 135 deg. F.

The cooling system should then be based on a pumping and cooling tower capacity of at least 60 lb. per brake horsepower per hour of installed engine rating. The gallons of water per horsepower hour for various temperature rises is shown in Table XXIV.

TABLE XXIV.—TABLE OF COOLING WATER REQUIRED IN GALLONS PER BRAKE HORSEPOWER HOUR

Inlet temperature	Discharge temperature, deg							
	95	100	105	110	115	120	125	120
50	7.2	6.5	5.9	5.4	5.0	4.6	4.3	4.1
55	8.1	7.2	6.5	5.9	5.4	5.0	4.6	4.3
60	9.2	8.1	7.2	6.5	5.9	5.4	5.0	4.6
65	10.8	9.2	8.1	7.2	6.5	5.9	5.4	5.0
70	13.0	10.8	9.2	8.1	7.2	6.5	5.9	5.4
75	16.2	13.0	10.8	9.2	8.1	7.2	6.5	5.9
80	21.6	16.2	13.0	10.8	9.2	8.1	7.2	6.5
85	32.4	21.6	16.2	13.0	10.8	9.2	8.1	7.2
90	....	32.4	21.6	16.2	13.0	10.9	9.2	8.1
			32.4	21.6	16.2	13.0	10.8	9.2

Types of Cooling Systems. Open System for Small Plants.—Two designs of cooling systems are in quite general use. Figure 280 outlines the open system often found in small installations. With this design the water from the engine jacket is discharged through a distributing pipe *D* on a cooling tower *C*. The water drips down through the tower and is stored in the sump *A*, from which point it is drawn by the circulating pump *B* and forced through the engine jacket and out the discharge again. When this system is adapted to a horizontal engine, the discharge line should rise vertically from the engine until it is above the cooling tower distributing pipe. With such a layout it is necessary to place a vent pipe in the discharge line, immediately above the engine. This prevents the formation of steam or air pockets in the jacket with consequent overheating of the cylinder.

While the figure embodies a centrifugal circulating sump, this design of pump is one that should never be employed with a

TABLE XXV.—TYPICAL JACKET-COOLING WATER DATA FOR DIESEL ENGINES

3-500 hp. McIntosh & Seymour Diesels, 164 r p.m.

Unit No. 3	Pounds per minute				Temperature (deg Fahr.)				B.t u. absorbed per minute			
	Full	¾	½	¼	Full	¾	½	¼	Full	¾	½	¼
Engine pump intake	.		...	...	83	83	83	77	.....	.....	......	......
Air-compressor discharge	48	33	...	44	110	110	101	104	1,796	891	......	1,188
Intercooler discharge	None in use											
After cooler discharge	31	33	....	36	106	120	136	110	713	1,221	......	1,188
Exhaust-header discharge	171	139	141	176	120	118	102	100	4,327	4,865	2,689	4,048
Cylinder No. 1: Jacket	52	48	42	32	138	132	131	132	2,860	2,352	2,016	1,760
Exhaust valve	11	8	9	10	102	100	96	92	209	136	117	150
Cylinder No. 2: Jacket	56	48	40	36	140	132	130	126	3,192	2,352	1,880	1,764
Exhaust valve	11	8	9	10	106	104	98	94	253	168	135	170
Cylinder No. 3: Jacket	55	48	43	34	140	132	127	126	3,135	2,352	1,892	1,666
Exhaust valve	12	8	10	11	90	104	96	92	84	168	130	165
Cylinder No. 4: Jacket	70	76	49	49	144	122	128	122	4,270	2,964	2,205	2,205
Exhaust valve	11	9	9	10	106	104	98	96	253	189	135	190
Total: Unit No. 3	528	458	352[1]	522	.		.	..	21,092	17,658	11,128[1]	14,494
Total: Unit No. 2	698 5	488	482	. . .	.		. . .	...	23,115	16,921	14,180	
Total: Unit No. 1	647	499	485		.	...	..	...	22,975	18,424	13,255	

[1] Except for air compressors.

system not having an overhead tank. The objection to this pump is based on the liability of losing the suction. This is of frequent occurrence when the drive belt becomes dirty or oily, and in the best-kept plants a belt at times breaks.

There is a serious objection to this system that makes its use inadvisable under any condition with a Diesel engine. The discharge, being at some distance from the engine, is not under the observation of the operator. The circulation can be broken without the knowledge of the operating force; this has resulted in broken cylinder heads and jackets in a number of cases. As a safeguard, a check valve with an outside lever connected to a bell-ringer is recommended.

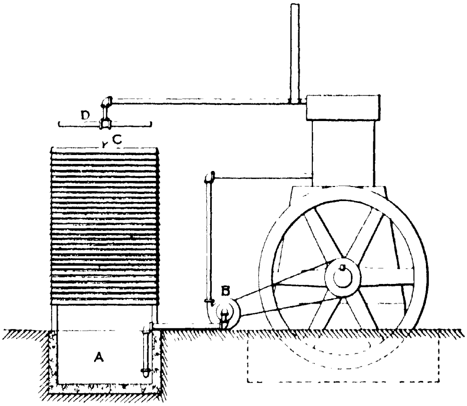


FIG. 280.—"Open" cooling water system.

Open Cooling System.—Figure 281 is the schematic layout of the open cooling system with a storage tank. With this system, the water is stored in the overhead tank and enters the cylinder jacket at *A*; after cooling the engine, the water discharges into the open funnel at *B*, flowing into the sump *C*. A centrifugal pump *D* lifts the water from this sump and discharges it in the top of the cooling tower *E*. In dripping down this tower the water is cooled and, collecting in the catch basin *F*, is lifted by the pump *G* and forced into the overhead tower. This system is frequently used without the overhead tank. With this plan the pump *G* forces the water through the engine jacket. It is at once apparent that this latter plan is objectionable since it is dependent on the pump *G* for the steady flow of water. If the suction is destroyed, the system at once becomes dangerous. Low-pressure engines are often installed with this system, but the cost of a Diesel plant is entirely too great to ignore the overhead tank. A 30,000-gal. tank with a 30-ft. steel tower can be installed at a cost of approximately $4,000. The interest on the investment ($240) is a low premium on the insurance of protection against engine damage due to lack of cooling water.

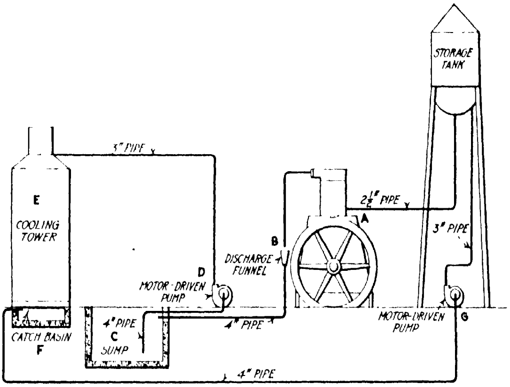


FIG. 281.—Open cooling system with overhead tank.

With a few engines the cooling water, after passing through the engine, is used to cool the exhaust header. With others the water is first passed around the air compressor and inter- and after-cooler before entering the engine jackets. Results from these cooling methods are fairly satisfactory. The proper cooling pipe layout embodies individual lines to the air compressor and coolers, to each engine cylinder jacket, to the valve cages and to the exhaust headers. All these lines should have brass cocks in the intake side, and the discharges should all lead to a common discharge funnel. Each discharge line may well be fitted with thermometers, while a single thermometer on the intake line before the lines branch to the various parts is sufficient.

Faults of Open Systems.—Not only must the supply of water be adequate to remove the heat rapidly from the cylinder walls, but the water must not contain much impurities either in solution or in suspension. If it does, the water jackets will soon become clogged with scale or mud, the effects of which are just as harmful in an oil-engine water jacket as in the water space of a steam boiler. However, there is this difference: In a steam boiler, scale will form from the sulphates and carbonates of calcium and magnesium held in solution, while in an engine water jacket, where the temperature never exceeds the boiling point (212 deg. F.), only the carbonates are precipitated as scale when the water is heated. In other words, a certain water containing a considerable quantity of sulphates but not carbonates might have a bad record as a boiler water, but would be satisfactory for cooling an engine. However, such cases are comparatively rare, as waters containing any mineral salts in solution are seldom free from carbonates. Water containing more than 6 grains per gallon of calcium and magnesium carbonates should not be used in engine jackets.

Carbonates are held in solution by an excess of carbon-dioxide gas in the water. When this gas is driven off by heat, the carbonates are precipitated in the form of a hard white scale which adheres firmly to the walls of the jacket and forms a non-conducting coating which retards the transmission of heat from the metal walls to the water. The consequence is that the metal walls attain an abnormally high temperature, causing inequalities in expansion and, when the scale is thick enough, finally resulting in excessive stresses which cause the casting to crack.

Originally, water cooling systems were designed merely for the purpose of saving water where it was scarce or expensive. More recently, however, such systems are being used, even though hard or dirty water is abundant, in order to supply engines with soft clean water.

The cooling towers, or open cooling systems, just described, have unfortunately two serious faults. In the first place there is a considerable loss of water, through evaporation and wind, and in arid climates where water is scarce, sufficient makeup water is often hard to obtain at a reasonable cost. The loss is usually from 4 to 10 per cent of the water passed over the cooling tower. Many oil engines are installed in arid districts in Arizona, Kansas, Oklahoma, etc., where this quantity of makeup water can be obtained only at prohibitive cost.

The second serious fault of open cooling systems is the fact that they concentrate the impurities in the water. If the water is hard, scale will be formed in the engine water jackets the same as in a simple cooling system where the water is discharged to waste. But in the open cooling system, even with water only slightly hard, concentration of the mineral salts in solution occurs rapidly, so that a water originally safe soon becomes loaded with scale-forming constituents and deposits a coating of scale in the engine water jackets. This concentration of impurities occurs because the evaporation at the cooling tower consists of pure water only and the materials originally dissolved in the evaporated water remain behind, dissolved in the rest of the water. With new makeup water mineral constituents are added to the system, which thus becomes more and more impure, resulting in scale deposits in the water jackets. The same process of concentration of impurities occurs in a steam boiler and necessitates blowing down occasionally to get rid of the impure water.

Because of these two serious faults of open cooling systems—expense of makeup water and danger of scale—the modern inclosed cooling systems were developed.

Inclosed Cooling Systems.—The modern cooling system goes under the name of Inclosed Cooling System. This type overcomes the two serious faults of open cooling systems; namely, expense of makeup water and danger of scale. It requires practically no makeup water and, when once filled with soft water which cannot form scale, will recirculate it continuously for a long period.

The principal feature is that the jacket water is recooled in pipe coils over which raw water is showered. The only place where the engine water is exposed is at the visible outlet. The loss is therefore negligible and a sufficient amount of good makeup water (rain water, for example) can almost always be easily obtained. Any quantity of water may be showered over the outside of the cooling coils.

A typical inclosed cooling system is shown in Fig. 282. This system is filled at the start with soft water. The warm water from the engine outlet is discharged into a funnel on the engine-room wall, from which the water runs into a sump pit. From this pit a circulating pump delivers the water into the cooling coils (usually outside the building) and thence up into the overhead tank from which the engine is supplied by gravity.

The raw water for cooling the pipe coils, if available in large quantities, may be distributed over the coils by means of a slotted gutter pipe in the same manner as an atmospheric ammonia condenser in a refrigerating system. With this arrangement a cooling coil made of twelve 2-in. pipes 20 ft. long is adequate for a 100-hp. modern-type oil engine. For an engine of this size the raw-water supply should be 25 gal. per minute. The capacity of the overhead storage tank, the sump pit and the soft-water circulating pump should be the same as for the open cooling systems previously described.

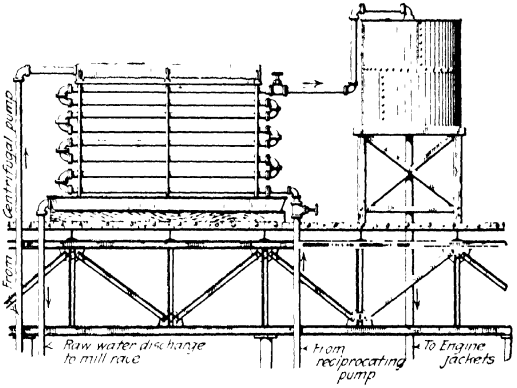


Fig. 282. Closed cooling system.

Where there is a scarcity even of inferior water for showering over the outside of the pipe coils the inclosed cooling system as arranged in Fig. 282 may be used. The raw water itself is caught in a basin below the pipes and pumped by a second pump over an open screen-type cooling tower, where it is recooled and then used again over the pipe coils. The logical location for the wire screens is over the pipe coils themselves, so that the cooled raw water as it drops off the bottom of the screens falls directly on the top pipes of the cooling coils. To secure the best results, the pipe coils are made lower than in the ordinary case previously described, and for a 100-hp. engine are recommended two coils each made of six 2-in. pipes 20 ft. long. Over each coil is mounted a $\frac{1}{4}$-in. mesh wire screen 6 ft. high and 20 ft. long.

A second enclosed system is illustrated in Fig. 283. This is the arrangement used on four 750-hp. Busch-Sulzer two-stroke-cycle engines in the Calumet sewer-pumping plant at Chicago, Ill.

The cooling system consists principally of a double centrifugal pump unit with a motor having its shaft extended at both ends mounted between them as a common prime mover, a heat exchanger and a booster centrifugal pump for handling the piston cooling water and necessary piping.

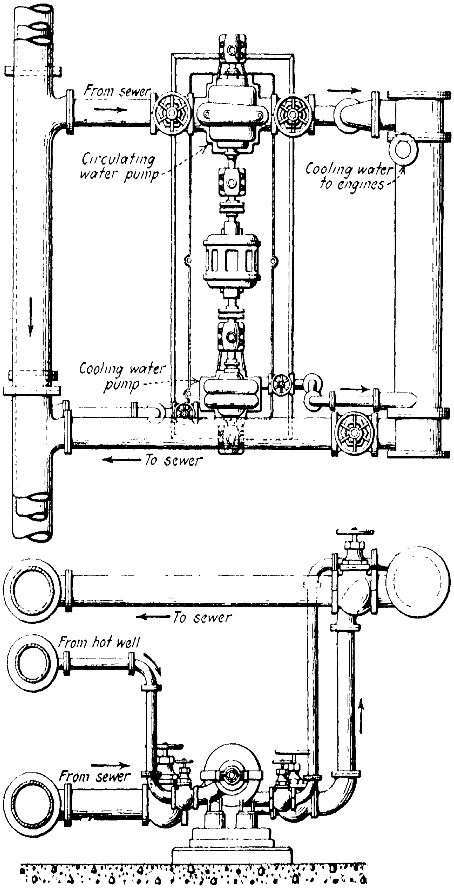


FIG. 283.—Cooling system with heat exchanger.

Circulating or raw water, which, normally, will be river water, is drawn from a storm sewer by the circulating-water pump, a basket strainer being placed in the suction line. The water is circulated through the tube side of the heat exchanger, Fig. 283, and out to the discharge sewer.

The engine-cooling water is taken from a hotwell in the basement by the cooling-water pump which is the second element of the double pumping unit, and is forced through the shell side of the heat exchanger, and then through the cooling-water passages of the engine. The cooling water upon leaving the engine flows by gravity back to the hotwell, where it is again circulated through the cooling system. A small booster centrifugal pump takes a part of the cooling water as it leaves the heat exchanger and forces it through the cooling passages of the engine pistons, the water returning to the hotwell by gravity.

Water Pipe.—All the water lines in the engine room are best laid in pipe chases. This places the piping out of sight and provides more room in the plant. The pipe should be extra heavy galvanized pipe; though the pressure is small, this thickness of pipe prolongs the life of the system. The threads in all fittings must be clean and sharp, while the pipe ends should be fully threaded. Red lead or other dope must be avoided, while the unions should have either ground joints or copper gaskets; rubber gaskets are at best of short life, and a pipe line should be made up in such a way that it will never give trouble. In those engines where water is admitted direct into the exhaust pipe, the drip line from the exhaust should not discharge into the cooling-water discharge; due to the carbon in suspension, it is advisable to run this drip to the sewer. Frequently the fuel contains enough sulphur to eat iron pipe if employed in the drip line; consequently brass drips can be more profitably used.

Cooling Towers and Tanks.—As has been already explained, an overhead tank is advisable in every Diesel installation. A steel tower and tank of from 25,000 to 30,000 gal. capacity is sufficient for any installation under 3,000 hp. since in this maximum case a 30,000-gal. tank would provide, in the event of pump failure, cooling water for one and one-half hours. Plants of 300 hp. or less will find a wooden tank on a wood tower quite satisfactory, Fig. 284. A 12 by 12-ft. tank of 2-in. cypress staves will hold close to 12,000 gal. and can be erected on a 30-ft. tower at a cost of $1,200. A tower made of 8 by 8-in. yellow pine with the joints reinforced by steel plates and braced by diagonal steel rods is amply strong for this tank.

Cooling Towers.—If a supply of raw cooling water can be secured from a shallow well, the best plan is to install a power well pump and allow the discharge to waste into the sewer. Unfortunately such a supply is seldom available, and some form of cooling tower whereby the discharge water can be cooled for reuse becomes a necessity. This, in most installations, consists of an upright wooden tower filled with slats down which the water trickles and is cooled by the upward current of air. Like all

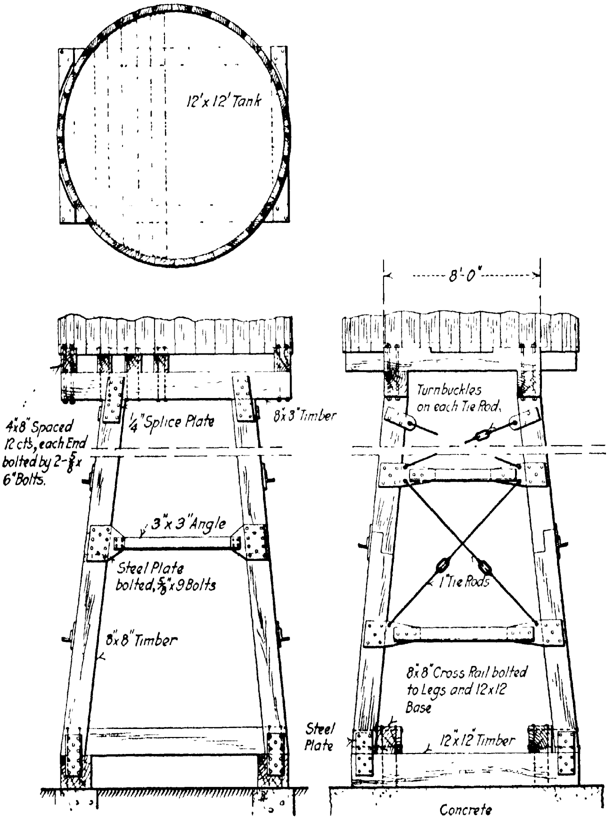

12'x12' Tank
8'-0"
4"x8" Spaced 12 cts, each End bolted by 2-$\frac{5}{8}$x 6" Bolts.
$\frac{1}{4}$" Splice Plate
8"x3" Timber
Turnbuckles on each Tie Rod
3"x3" Angle
Steel Plate bolted, $\frac{5}{8}$"x9 Bolts
1" Tie Rods
8"x8" Timber
8"x8" Cross Rail bolted to Legs and 12x12 Base
Steel Plate
12"x12" Timber
Concrete

FIG. 284.—Wooden tank and tower.

structures, the tower may be constructed at an expense ranging from a few hundred to several thousand dollars. The former cost covers a simple tower for a small plant, while a large installation demands the more expensive construction.

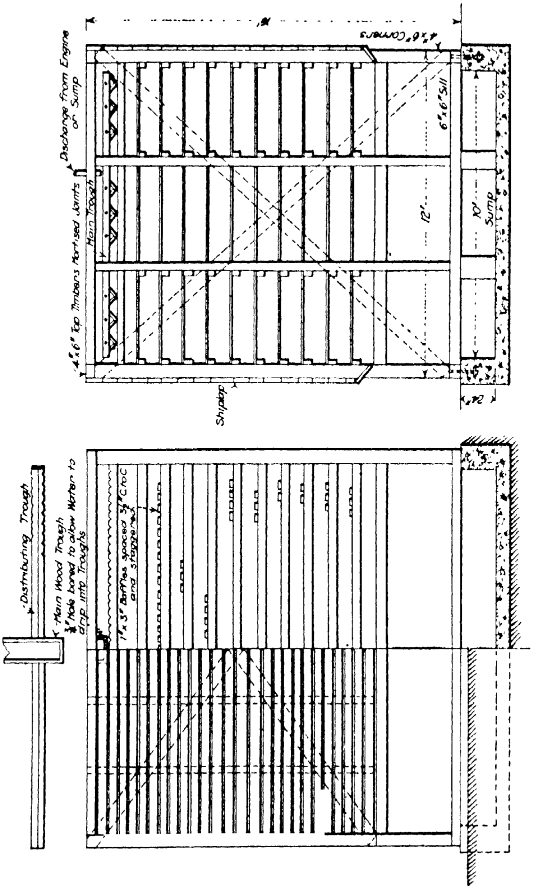

Discharge from Engine or Sump
4"x6" Top Timbers Mortised Joints
Main Trough
4"x6" Corners
6"x6" Sill
12'
10' Sump
12"
Shiplap
Distributing Trough
Main Wood Trough
½" hole bored to allow Water to drip into Troughs
1"x3" Baffles spaced 3" C to C and staggered

FIG. 285.—Cooling tower for 300 hp. plant.

Plants of 300 hp. will find the tower in Fig. 285 fairly economical in first cost and amply large for the required cooling. This tower can be erected at a total material and labor charge of $600. The sump under a tower is made 24 in. deep but can easily be deepened at slight expense. The ¾ by 16-in. foundation bolts should be inserted at the time the concrete is poured. The frame

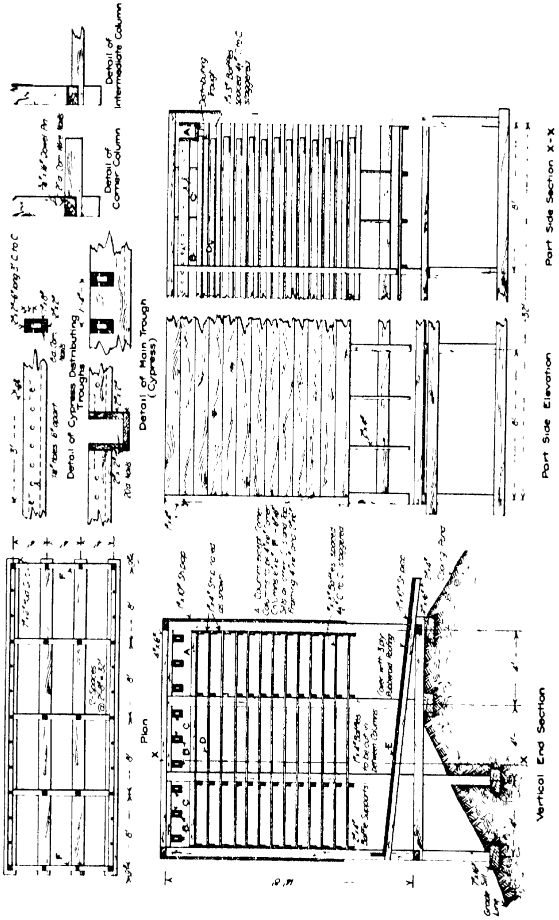


FIG. 286.—Cooling tower 1,000 hp. plant.

of 4 by 6-in. timbers, resting on 6 by 6-in. sills, is large enough, especially since the ship-lap sides further strengthen the structure. The sides are sealed with the ship-lap with the idea of having the air currents enter the tower under the bottom row of baffles and pass out at the top. If the sides are open, practically no circulation is set up.

The discharge from the engine first flows into the main trough and, passing through a series of holes in the trough sides, enters the distributing troughs. The latter are notched to allow the water to overflow before completely filling the trough. This feature is of advantage when the troughs settle. The holes in the main trough can be plugged, thereby offering means of controlling the distribution of the water.

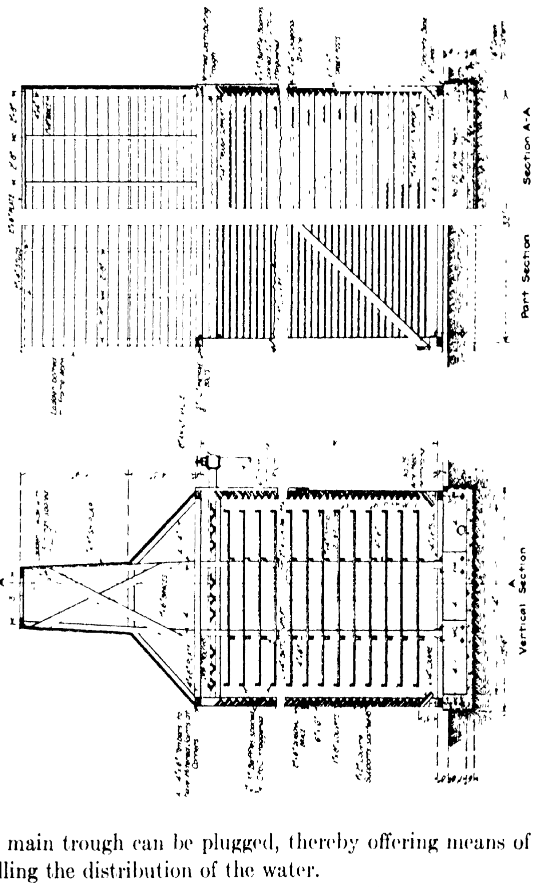


FIG. 286A.—Cooling tower, 1,500 hp. plant.

Figure 286 shows a somewhat similar design for an installation of 1,000 hp. In the particular plant where this tower was used a cooling pond with spray nozzles was originally employed. In erecting this tower, the pond was retained as the sump. As the drawing shows, the tower was placed on the pond bank, and a drip surface *E* of 1 by 10-in. ship-lap covered with three-ply rubberoid roofing was placed under the tower. In this drawing the tower sides are boarded tight with ship-lap However, this was not carried down below the bottom baffles. On windy days the water was blown out the open sides at the bottom, causing a washing of the bank and a settling of the tower sills. In erecting a tower, the sides should always be carried well down below the bottom baffles.

For plants having a capacity of 1,500 to 2,000 hp. the more elaborate tower design in Fig. 286*A* will prove more suitable. This tower has the sides covered with louvres while the top is carried above the trough, forming a chimney. This construction provides an increased volume of air passing through the sides while the chimney gives the necessary draft.

Circulating Pumps.—In the small one-engine plants where a single operator cares for the machinery, the most approved type of circulating pump is either an outside packed triplex pump or a horizontal plunger pump. A large percentage of plants have centrifugal pumps, but this type is dangerous in the small plant. If an overhead tank is a part of the installation, the centrifugal pump offers no objectionable feature. In those plants not possessing the protection of the overhead tank, the loss of the pump suction, as often occurs with this pump, is dangerous. The pump, in small plants, should, if possible, be chain-driven from the engine shaft since a broken belt is to be avoided at all costs.

Large installations usually employ motor-driven centrifugal pumps. With the overhead storage as a plant protection, this pump is by far the best for large plants.

In placing the pump, the suction line should be made as short as possible and the lift kept at a low value. In discharging into the overhead tank the pipe should run to the top of the tank rather than enter the tank bottom. This places a constant head on the pump, and, in case the pump suction valve leaks the entire tank will not empty. A by-pass line from the tank bottom can be run into the pump discharge for priming purposes.

Both the tank and the cooling-tower pumps should be installed in accessible places in the plant. The common practice of putting the pumps in out-of-the-way corners has nothing to commend it. The various water lines can be painted in different colors;

this enables the operator to trace out a line with a minimum of difficulty.

Handling of Cooling Water.—After an engine is stopped, the working parts contain a large amount of heat. To remove this heat and bring the piston, cylinder, cylinder head to a low temperature without danger of any part cooling faster than another and so causing distortion, the cooling water should be allowed to run for from fifteen minutes to one-half hour. The flow need not be heavy.

Upon starting the unit the cooling water should be started through the engine before it is turned over. However, in engines that are worn considerably the cooling water may lower the cylinder temperature below the ignition point. Under such circumstances it is permissible to start the engine before the water valve is opened. Since the jacket is filled with warm water there will be no danger of fracture.

In extreme cases of low compression operators have gone so far as to drain the jackets before starting in order to eliminate all cooling loss. In one motorship this was the common practice, but the danger that accompanies the action is obvious. It is far better to have a small vertical boiler and heat the jacket water before starting the engine; even live steam may be used if the jackets are first heated with the steam and then filled with hot water.

The operator should endeavor to keep a constant jacket discharge temperature. Much damage has occurred by reason of too wide a variation in the jacket temperature.

Exhaust Distiller.—The logical method of water purification, if rain water is not available, is the employment of an exhaust distiller. This heater is placed in the engine exhaust line and absorbs part of the heat contained in the exhaust gases. The engine cooling water ordinarily loses 3 per cent of its volume in being cooled in case an open-system cooling tower is used. The make-up water necessary to balance this loss is first circulated through the heater and is converted into steam at atmospheric pressure. This steam is then led to a closed heater where it is condensed by cooling water from the source of supply. The heater is, in form, nothing but a steel tank filled with 2-in. tubes through which the water circulates. The heater in Fig. 287 has both heads flanged, which feature allows the tubes to be bored out with a hydraulic turbine tube cleaner.

To raise the temperature of the feed water before it enters the tank, the feed line can easily be passed through the engine exhaust pot. This will raise the temperature to about 150 deg. In computing the size of heater required for any given plant, 1 sq. ft. of heater surface per horsepower will be amply large to absorb all available exhaust heat. The heat wasted by the engine will average 3,000 B.t.u. per horsepower, and the heater will abstract 2,000 B.t.u. This will give roughly 2 lb. of make-up water per horsepower. Since the cooling loss will not exceed 3 per cent, this amount of make-up water is sufficient even when the quantity of water used exceeds the usual 60 lb. per brake horsepower.

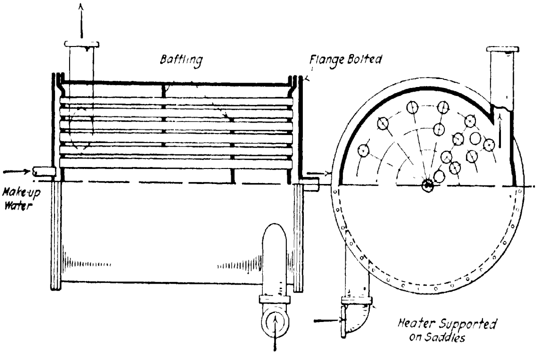


FIG. 287.—Make-up water distillery.

Temperature of Cooling Lines.—Each engine possesses individual characteristics that preclude any set rules as to the temperatures that should exist in the discharge cooling lines from the various parts. Table XXVI is the schedule that is followed in a plant containing three 500-hp. vertical Diesels. These values give the best possible operating results as applied to these particular units.

TABLE XXVI.—COOLING WATER DISCHARGE TEMPERATURES

	DEG. F
Air compressor discharge	105
Inter-and aftercooler discharge	105
Exhaust header discharge	130
Cylinder jacket discharge	150-160
Exhaust valve discharge	120

Recovering Heat in Cooling Water and Exhaust.—There is, as indicated in Fig. 279, a large amount of the heat of the fuel lost to the cooling-water jacket and to the exhaust. The sum of these two losses may be taken, strictly speaking, as 60 per cent of the heat in the fuel. By suitable means a large percentage of this loss may be recovered.

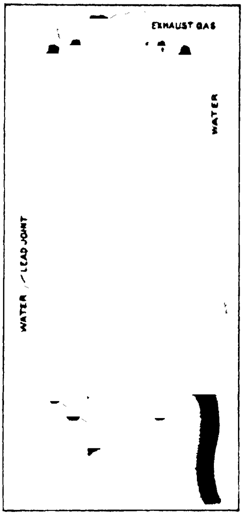


Fig. 287A. Sims exhaust heater

Two methods have been used with success. One of these, by far the simpler, is to connect the engine's cooling jacket to the heating system of the factory. By using ample radiating surface the entire jacket-water heat may be used in heating the plant. As example, assume the engine capacity is 500 hp. With a fuel consumption of 0.4 lb. per b.hp.-hr. the hourly rate of fuel consumption would be 500 × 0.4 = 200 lb. If the oil contains 18,500 B.t.u. per pound and the jacket absorbs 30 per cent of this, the jacket water will deliver hourly to the heating system 200 × 18,500 × 0.30 = 1,110,000 B.t.u. or the equivalent of approximately 1,100 lb. of steam. An evaporation of 5 lb. of water per pound of coal is representative of heating-plant practice. The jacket-water recovery is then equal to 220 lb. of coal per hour or 5,280 lb. per day.

Recovery of Exhaust Heat.—A large part of the exhaust-heat loss may be recovered by installing some form of exhaust gas heater. For low-pressure work, say 10 lb. gage, a castiron heat interchanger similar to Fig. 287*A* has been quite generally used in this country. The amount of steam evaporated, or water

TABLE XXVII.—CAPACITY OF SIMS HEATERS

Engine Horse Power	Dimensions of Heater	Number of Section	Water Connections	Exhaust Connections	Heating Capacity in gallons per hour using Exhaust only			Heating Capacity in gallons per hour using Exhaust and Jacket Water			Weight
					150°	180°	200°	150°	180°	200°	
10	18″x15″	2	1″	3″	45	35	30	110	80	70	360
15	18″x20″	3	1″	3″	70	50	45	160	125	120	475
20	18″x24″	4	1½″	5″	90	70	60	215	160	170	600
25	18″x30″	5	1½″	5″	110	85	75	270	210	180	700
30	18″x35″	6	1½″	5″	130	105	90	330	250	220	820
35	18″x40″	7	1½″	5″	150	120	105	380	290	250	935
40	18″x45″	8	1½″	5″	170	140	125	425	335	290	1050
45	18″x50″	9	1½″	5″	190	160	140	490	375	330	1160
50	18″x55″	10	1½″	5″	210	175	155	545	420	370	1280
60	24″x39″	6	2″	8″	250	200	175	635	485	430	1640
70	24″x44″	7	2″	8″	290	235	200	720	550	490	1860
75	24″x49″	8	2″	8″	340	260	230	810	620	550	2080
100	30″x39″	5	2″	8″	450	350	300	1090	840	720	2220
125	30″x45″	6	2″	8″	580	430	380	1360	1040	900	2570
150	30″x51″	7	2″	8″	680	520	460	1630	1250	1100	2900
175	30″x57″	8	2½″	12″	780	610	535	1840	1420	1270	3260
200	30″x64″	9	2½″	12″	900	700	600	2050	1670	1450	3600
225	30″x70″	10	2½″	12″	1000	780	690	2390	1880	1640	3950
250	30″x77″	11	2½″	12″	1150	870	760	2720	2100	1820	4300
275	30″x83″	12	2½″	12″	1250	960	840	2900	2300	2000	4650
300	30″x90″	13	2½″	12″	1350	1050	920	3140	2500	2180	5000

Using both Exhaust and Jacket Water, this Heater will produce 5 lbs. of Steam per H. P. per hour at 10 lbs. pressure.

heated, is shown in Table XXVII, this data being issued by the Sims Heater Co.

For high-pressure work a steel shell boiler is necessary and the length of the boiler must be such that the gas temperature does not drop below the vapor condensation point. If this is not taken care of, corrosion will set in in the tube ends.

The waste-heat boiler has been largely developed in England, and if part of the jacket water is used for the boiler feed it is possible to obtain a boiler output of from 2 to 3 lb. of steam per Diesel horsepower output.

CHAPTER XVIII

DIESEL FUEL OILS

COMBUSTION. CLASSIFICATION OF FUELS. ACTION OF FUELS. FUEL SPECIFICATIONS. FILTERS. TANKS

Combustion.—The process of combustion which takes place in the cylinder of an internal-combustion engine is simply a chemical reaction. In actuality the cylinder is merely a chemist's retort wherein the atoms of hydrogen and carbon, which make up the body of the fuel charge, unite with the oxygen contained in the air charge, forming oxides. The carbon in its union with oxygen forms either carbon monoxide (CO) or carbon dioxide (CO_2). In this latter chemical reaction an atom of carbon unites with two atoms of oxygen forming one molecule of carbon dioxide; this combustion releases 14,600 B.t.u. per pound of carbon and raises the remaining unburnt carbon to the incandescent point. Unless there is sufficient oxygen present to unite with this incandescent carbon, the latter unites with one of the oxygen atoms of the carbon dioxide (CO_2), causing the entire carbon oxide to assume the form of carbon monoxide (CO). Since the reaction is incomplete, the heat released by the formation of carbon monoxide from carbon and oxygen is less than that produced by the complete reaction, CO_2. The value is approximately 4,380 B.t.u., making evident the heat loss when the combustion is not complete. All Diesel cylinders are of ample volume to give sufficient oxygen for complete combustion. If the chemical reaction is not fully carried out, it is due to causes other than an insufficient supply of oxygen. In most instances the defect is traceable to poor atomization wherein the oil charge is not separated into particles of such minute dimensions that each carbon atom contacts with the required oxygen atoms. If the oil droplets entering the cylinder are of fairly large diameter, the oxygen is in direct contact with only the carbon at the surface of the droplet. The carbon atoms within the droplet must receive their oxygen from the carbon dioxide formed at the surface, causing the CO_2 to break into CO and O_2 to supply oxygen to the inner carbon atom. For this reason the engine's efficiency depends on the degree of atomization of the fuel charge. The chemical reactions taking place probably follow this order:

$$C + O_2 = CO_2 \quad (1)$$
$$C + CO_2 = 2CO \quad (2)$$
$$2CO + O_2 = 2CO_2 \quad (3)$$

The hydrogen atom of the oil molecule also unites with the oxygen, forming water (H_2O) or rather water vapor commonly called steam, the reaction being as follows:

$$2H_2 + O_2 = 2H_2O \quad (4)$$

This reaction generates 62,100 B.t.u. per pound of hydrogen. The equations do not refer to the actual weight of the carbon, oxygen and hydrogen but merely indicate the relation of the atoms. Since the atomic weights of the various substances differ, it follows that the weight of each substance entering into the reaction depends on its atomic weight wherein a hydrogen atom has unity weight, carbon 12 and oxygen 16. Furthermore, where oxygen and hydrogen are not in combination with other gases, both oxygen and hydrogen have their atoms, or most minute particles, in groups of two or more. Equation (1) can then be written

$$C + O_2 = CO_2$$
$$12 + 32 = 44$$

indicating that 12 parts by weight of carbon combining with 32 parts of oxygen form 44 parts of carbon dioxide; then, 1 lb. of carbon requires $^{32}/_{12}$ or 2.67 lb. of oxygen to be converted into 3.67 lb. of CO_2. Since a pound of air contains $^{23}/_{100}$ lb. of oxygen, there are 11.6 lb. of air required in the combustion of 1 lb. of carbon. Air of 62 deg. F. has a volume of 13.14 cu. ft. per pound; the 11.6 lb. then have a volume of 152.4 cu. ft. The hydrogen reaction (4) can be written as follows:

$$2H_2 + O_2 = 2H_2O$$
$$4 + 32 = 36$$

where 1 lb. of hydrogen requires 8 lb. of oxygen or 34.78 lb. of air.

While the petroleum oils contain hydrocarbons of a varied structure, the equation below, covering the ethylene series of hydrocarbons, outlines the process of equating the reactions taking place.

$$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$$
$$28 + 96 = 88 + 36$$

where 28 lb. of ethylene require 96 lb. of oxygen to form 88 lb. of carbon dioxide and 36 lb. of water or steam. Since 23 per cent of the air is oxygen, 417 lb. of air are required to consume the 28 lb. of C_2H_4, of which 77 per cent, or 321 lb., is nitrogen, which experiences no chemical reaction. Then, for perfect combustion, the percentage by weight of the exhaust products would be

CO_2 Carbon dioxide	H_2O Water	N Nitrogen
20	8	72

These are the theoretical percentages and are quite different from those obtained on an actual Diesel engine test where the percentages obtained were as follows:

CO_2 Carbon dioxide	CO Carbon monoxide	N Nitrogen	O Oxygen
7.2	0.2	81.6	11

The H_2O percentage was not obtained in the test.

In Diesel construction the designer always provides a cylinder volume which gives a large excess air capacity. It is usual to figure on from 18 to 28 lb. of air per lb. of fuel.

Fuel Classification.—Liquid fuels that are available for use in the Diesel engines fall into three groups. The first includes those hydrocarbons of the paraffine, olefiant and acetylene series, such as are familiarly known as the petroleum oils, the principal family being the paraffine series C_nH_{2n+2}. The second group covers the liquids known as benzol derivatives, such as coal tars. The third group is composed of the many vegetable oils containing hydrocarbons, such as peanut oil, palm oil, cocoanut oil and oils from other tropical and sub-tropical vegetation. The last group, up to the present time, has not been used commercially since their cost has been far above that of the petroleums. The coal tars are employed to a considerable degree in Europe; Germany, especially, has given much attention to the adaptation of the Diesel to these oils. The increasing favor of the tar oils has been largely induced by the low cost of these oils and by the import duty laid on the petroleum oils. In Germany the import duty is equal to the cost of the fuel delivered at the wharf.

In the United States no fuel other than the petroleum oils has found favor; this is attributable to the low price and convenience of these oils. Some speculation has been indulged in as to the advisability of using the American tar oils. However, this will be unnecessary for many years to come since the petroleum output is about 500,000,000 barrels per year. Approximately 40 per cent, or 200,000,000 barrels, of this oil remains after the gasolene, kerosene and light distillates have been distilled off. These heavy residual oils are marketable as road surfacers, boiler fuel and Diesel fuel oils. Since the Diesel plant can offer a higher price than is justified by the boiler plant, a large part of this oil may be regarded as a potential Diesel supply. The total of 200,000,000 barrels would furnish fuel for more than 15,000,000 hp. on a 24-hr. service. This is many times the total Diesel installation in America.

Furthermore, the Mexican oil fields are of vast productive possibilities. The many internal disturbances in Mexico have served somewhat to limit the output; this is being overcome, and large quantities of Mexican crudes are even now on the market. These oils offer but small inducements to the refiner since they carry very low percentages of the lighter distillates. The Mexican oils are marketable only as boiler fuels and Diesel oils. Their employment in a Diesel entails more attention from the engine operator than do the higher gravity oils of the United States. There is however, no constructional grounds for ignoring the Mexican fuels when the supply of the American oils is depleted.

Of the petroleum oils of the United States those of the Eastern and Midcontinental fields prove the most satisfactory in Diesel operation. This is a generality, however, and oils from wells in the Southern Texas fields, where the fuel is somewhat similar to Mexican crudes, give excellent results. The Diesel plant can purchase oils covered by rigid specifications, but the only positive way to determine the suitability of an oil is to run a test. It is of no advantage to test a single barrel since no engine will reveal the objectionable features of a fuel on a one-barrel test. The lowest quantity to experiment with is at least ten barrels.

The petroleum oils on the market as Diesel fuels range from South Texas and Mexican crudes, which contain too small a percentage of gasoline to justify distillation, up to 40 deg. distillate, which is actually a low-grade kerosene. Most Diesels are burning fuel oil from 20 to 30 deg. Bé., which is a residue left after the gasoline and kerosene constituents have been distilled out of the crude and which has been filtered to remove the dirt. This oil ordinarily is desulphurized, which process eliminates practically all the sulphuric acid which was introduced in the distillation. This fuel oil has a varied color, ranging from light yellow to a deep black. The color is no criterion of the gravity of the oil, the crudes of various fields differing widely in color.

The Diesel fuels fall under the following grades: Stove Oil, Solar Oil, Gas Oil, Distillate, Fuel Oil, Diesel Oil, Desulphurized Fuel Oil, Crude Oil, Tops.

Stove Oil.—This is a trade name attached to a low-grade kerosene from 39 to 44 deg. Bé. gravity. The oil is ordinarily of a yellow-white color having a dirty appearance. It is quite serviceable for a low-compression engine but cannot be termed a desirable Diesel oil since it flows very readily and will pass the injection atomizer disks without being broken up. Blowing into the cylinder ahead of the air, it produces severe preignitions. It is of advantage only as a starting oil that can be used in cold weather for the purpose of warming the engine before the heavier fuel oil is introduced. It constitutes a serious fire hazard if any large quantity is kept on hand. Its price, which averages 2 to 4 cents above fuel oil, would make its adoption inadvisable even if it were an otherwise desirable fuel.

Solar Oil.—This is a trade name covering some of the lighter grade distillates and may vary over a wide range in gravity from 36 to 42 deg. Bé. Like stove oil, it cannot be classed among the desirable fuels and should be used only where frequent starting and stopping are usual, such as in a Diesel-engined tug. Serious preignitions are of frequent occurrence with this fuel. Its price is usually some 2 cents per gallon above fuel-oil quotations.

Fuel Oil.—This term covers all oil residue that remains after distillation has removed the gasoline, kerosene and light distillates. Dependent on the particular crude, the fuel oil may vary in gravity from 20 to 32 deg. Bé. In practically all cases it is black in color, though some of the Eastern oils have a light color, as also have certain North Texas fuel oils. When properly filtered, etc., it is the most desirable fuel that can be purchased. Since all the light volatile gases have been removed, there is little danger. It is low in price in comparison with either crude or the light oils.

Gas Oil.—After the kerosene, gasoline, etc., have been removed, if the distillation temperature is sufficiently increased, all the remainder of the crude will distil over, with the exception of a heavy tarry portion which is sold for road surfacing. These heavy distillations are called gas oils and are sold for Diesel consumption. They are superior to the fuel oils, being free from asphaltum residue, but their price, which is above that of fuel oil, somewhat limits their application.

Diesel Oil.—This is a refinery trade name for a filter fuel oil. It is sold at a slight advance over the unfiltered fuel and has a ready market in the Midwest. For small plants where no filtering apparatus is installed it is the fuel that should be purchased.

Distillate Oil.—The distillate oils range in gravity from 30 to 39 deg. Bé. They have a yellow-green tinge and are the products of a distillation, being vaporized at a higher temperature than is kerosene or stove oil. They are ideal fuels for hot-bulb engines, but the price is entirely too high to be attractive for Diesel use.

Tops.—Topped oil is the residue after the gasolene has been removed. It is not commercially offered in any but limited quantities and can be ignored in making a test of Diesel fuels.

Crude Oil.—Within this class fall all crude oils which have undergone no process of distillation and which are marketed as they come from the well. With the exception of some South Texas and Mexican crudes which do not justify distillation, no crude can be obtained save in limited quantities. In some localities the owners of small wells sell the raw crude to the local plants. Since the range in the character of oils forming the crude is great, it is not a desirable oil. The gasolene content becomes a fire hazard that cannot be ignored. The functioning of the engine on the crude is by no means perfect. The Mexican crudes do not carry such a range in hydrocarbons and perform fairly satisfactorily if the engine's fuel nozzle and combustion chamber are designed to atomize and mix the air and fuel in an efficient manner.

Using Low Grade Oil in the Diesel Engine.—The first problems that arise in connection with the use of the low-grade fuels are those involved in transportation, storage and pumping, due to their high viscosity; and since viscosity is a function of temperature, these problems concern methods of heat application. It should be noted that the relation between Baumé gravity and viscosity is not constant, and two oils from different localities may have the same Baumé gravity, yet vary considerably in viscosity. For this reason the fact that an engine is operating on 16-deg. Bé. fuel oil does not necessarily mean that the same method of treatment would permit operation with another oil of the same gravity. The Baumé gravity does, however, give a rough indication of the suitability of an oil for use as Diesel-engine fuel and the conditioning treatment required.

As indicated by the low sulphur content called for in the specifications for special Diesel fuel, it has been considered that sulphur in the low-grade oils, resulting in the formation of sulphurous-acid gas during combustion, constituted the chief bar to their use. Water vapor is produced by the combustion of the hydrogen in the fuel, and if the exhaust gases are cooled sufficiently to condense this water vapor, it combines with the sulphurous-acid gas and becomes actively corrosive to the metal of the valves and cylinder liners. This condensation cannot occur while the engine is in operation because the exhaust gases leave the cylinders at a temperature of about 600 deg. F., which is much above the condensing temperature of the water vapor. The corrosive effect, then, can be expected only where the engine is stopped and after the gases left in the cylinders and exhaust passages cool to the condensation point. The easiest method of preventing this corrosion is to keep on hand a small quantity of the higher-grade Diesel oil or kerosene and use it for the engine fuel a few minutes before shutting down and likewise upon starting. This eliminates the danger of sulphurous-acid gas remaining in the cylinder during the cooling-down period and its formation during the warming-up period, for kerosene contains no sulphur to unite with the water vapor. This method of operation is desirable for other reasons to be pointed out later.

Practically all the available boiler oils are asphalt-base oils and the principal difficulty in their use arises from the asphalt content, which is much harder to burn than any of the other constituents. If conditions are not right, an engine may operate for a considerable length of time by burning the more easily combustible constituents, leaving the asphaltic residue to collect in the piston-ring slots, on valve stems and in exhaust passages in the form of a tough, elastic gum. This gummy substance is combustible if sufficient heat is supplied, but after it once becomes attached to metal surfaces not directly in the zone of highest heat, the metal carries away the heat so rapidly that a temperature high enough to insure combustion of the gum cannot be attained. It can be removed only by opening up the cylinder and taking out the piston. This means that if operated under the condition just mentioned, the friction load due to the accumulation on the piston gradually reduces the effective power of the engine and finally forces a shutdown for removal and cleaning of piston and rings.

A cold piston may be removed from its cylinder so gummed up with this residue that the rings are frozen absolutely tight in their slots in the compressed position, but if the piston and rings are heated with a blowtorch, the residue becomes fluid and the rings will spring out to their normal position and can readily be removed. Cases have been known where this tenacious asphaltic gum has accumulated on the piston to such an extent that, with the cylinder head off and the turning gear in operation, the friction between piston and cylinder has been sufficient to push the liner out of the cylinder. In those cases it was noted that all the accumulation was on the sides of the pistons and the piston tops were quite clean. The top is normally the hottest part of the piston and there is sufficient heat to burn all the fuel that may strike the top.

Burning Crude Oil.—If the cylinders, heads, pistons and valves can be brought to normal working temperature before the crude oil is admitted, combustion will be complete and no residue will be deposited. It is practically impossible to heat these parts up to the desired temperature without actual running, and if the engine is started with the heavy fuel, several minutes' running must elapse before the normal running temperature is reached. During this time the unburned part of the fuel is depositing on the exposed metal surfaces, and even after the highest practicable working temperature is reached, this deposit cannot be burned off except from the surfaces within the combustion space. That part of it that reaches the sides of the piston and the piston-ring slots never burns. Every time the engine is started, the accumulation at these points is added to, and finally a condition is reached that makes starting impossible. During the period of incomplete combustion unburned particles of fuel are carried by the exhaust gases, and some of them strike the exposed portion of the exhaust-valve stem every time the valve opens. They accumulate in the clearance space between the valve stem and its guide and eventually cause the stem to stick. The simplest method of preventing these troubles is to start the engine on a light, easily burned fuel oil and operate with this oil until the engine is thoroughly heated, then switch to the heavy oil.

It is highly important, however, to maintain always this high-temperature condition within the cylinders. It is the highly heated compressed air within the combustion space that must be depended upon to burn the asphalt in the fuel. The temperature of this air depends upon two things—the amount it is compressed, or the ratio of its volume after compression to its volume before compression, and its temperature at the beginning of compression. For normal operation with the lighter grades of oil, the clearance space is adjusted to produce a pressure of 480 to 500 lb. of air after compression. To insure complete burning when using heavy oil, this compression pressure should be increased to 500 to 525 lb., and the jacket cooling water should be regulated so that the engine normally runs hotter with heavy fuel than with light. No fixed rule as to water temperature can be given that will be applicable to all engines, but usually the limiting factor is the clearance between cylinder walls and piston. Any jacket-water temperature below the boiling point is permissible if this clearance is sufficient to allow the resulting piston expansion.

Quite as important is the problem of keeping the fuel in a sufficiently fluid state to permit correct atomization in the cylinders and efficient performance of the fuel-measuring pumps. The degree of fineness to which the individual particles of oil are reduced in the atomizing process naturally has an influence on the ease of burning, as the more completely the oil is broken up the easier it is for the air and oil to combine. Viscosity has a direct effect on atomization and on the ability of the pumps to handle the oil, but it has been determined experimentally that if the viscosity of any oil is reduced sufficiently to meet the atomization requirements, it will be sufficiently fluid to pass through the pumps. The only practicable method for reducing viscosity is by the application of heat, and the amount of heat required varies with different oils.

Heating Fuel Oil.—It is frequently necessary in northern states to heat the oil before it will flow from the tank car to the storage tank. For this a steam coil supplied by a small boiler may be used. Hot water will seldom suffice since the tank coil is limited in size.

The fuel should be filtered by some form of centrifugal separator placed between the car and the storage as in Fig. 288. In this way the dirt and asphaltic substances never reach the storage and the latter is kept clean.

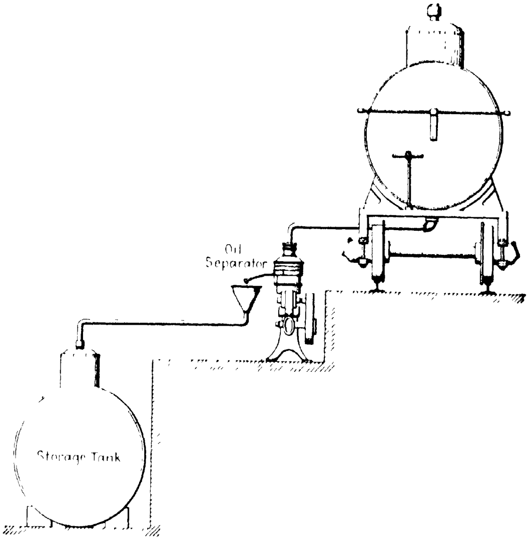


FIG. 288. Filtering oil into storage tank.

The storage tank should have a heating coil placed about the pump-suction pipe; hot water may be obtained in any of several ways. Such a coil is shown in Fig. 289. If the coil is placed along the bottom of the tank, 1 sq. ft. of pipe surface to each 60 ft. of tank volume is ample.

An effective way to obtain hot water for oil heating is to place a hot water heater in the exhaust line and run part of the warm jacket cooling water through this heater as outlined in Fig. 290.

In Fig. 291 the exhaust pipe has a copper coil inserted in one section. This coil is filled with water which circulates around the strainer shown just above the pumps.

Often a water-jacketed oil filter is all that is needed. Such an apparatus is illustrated in Fig. 291*A*, warm cylinder-jacket water being used to keep the oil fluid.

Boiler oil, as has been indicated, requires more operating care and adjustment. Consequently, before purchasing such fuel in preference to a, say, 20 deg. Bé. oil, these facts must be considered. Unless the fuel saving is more than enough to cover a large increase in maintenance cost the higher price fuel is to be preferred.

Heat Value of Fuel Oils.—It is self-evident that the oils having a higher heat value are more valuable than those oils with a low heating value. All other qualities being equal, the comparative values of two oils are in direct proportion with the ratio of their heating values. If oil of 36 deg. Bé. gravity having 18,000 B.t.u. per pound is quoted at $3.60 per 100 gal. or $5.12 per 1,000 lb., an oil containing 20,000 B.t.u. is equally attractive at $4 per 100 gal. or $5.69 per 1,000 lb. The policy of purchasing oil by the gallon is not to be recommended, although it is the custom to quote in this manner. Orders for oil should be placed on the pound basis since the heating value of oil is so computed. Since the pounds per gallon of fuel oil of 30 and 20 deg. Bé. are respectively 7.294 lb. and 7.777 lb., it is apparent why the heavier oil is worth more per gallon.

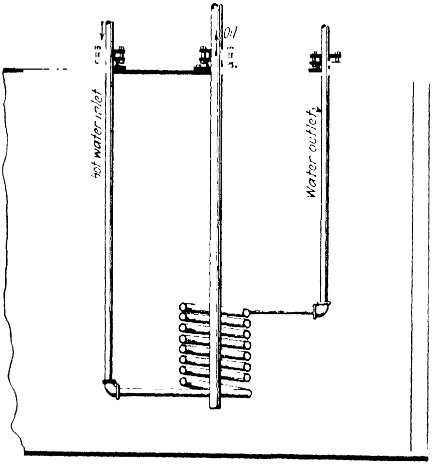


FIG. 289.—Tank heating coils

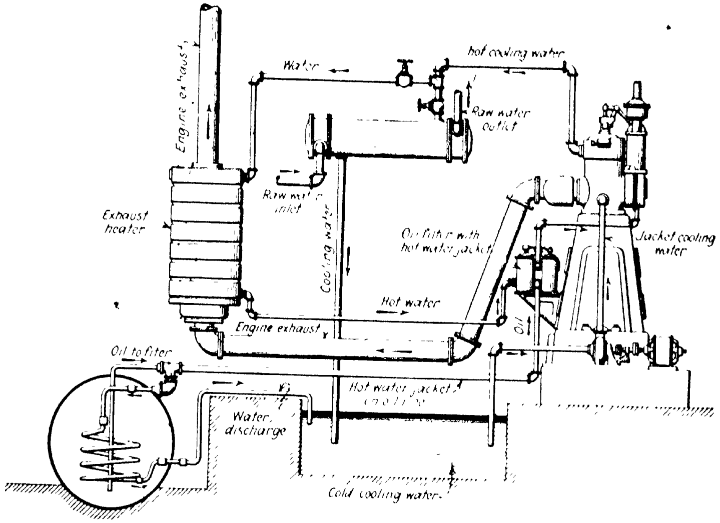


FIG 290 —Using jacket water to heat the fuel

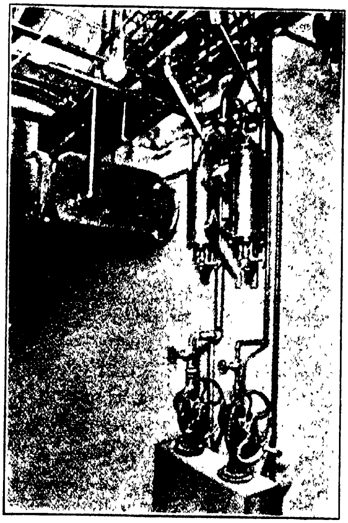


FIG. 291.—Heating oil by exhaust gases.

In purchasing oil the heating value should always be given or required. If the chemical analysis of the oil is given the heating value of fuel oil in B.t.u. per pound can be determined by Inchley's empirical formula:

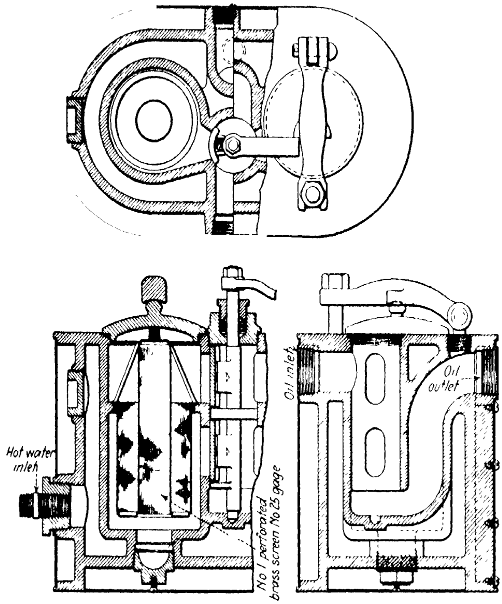


FIG. 291*A*.—Water-jacketed oil filter.

$$\text{B.t.u. per lb.} = 13{,}500\ C + 60{,}890\ H$$

in which C and H are the proportions by weight of carbon and hydrogen, respectively, and the heat value is the "high value." This is about 1,000 B.t.u. per pound higher than the low heat value, wherein an allowance is made for the total heat carried away from the furnace by the water vapor produced by the combustion of the hydrogen. To determine the lower heat value accurately, deduct from the higher value 8,730H in which H is the proportion by weight of hydrogen in the oil.

Another formula for the high value, proposed by Sherman and Kropff, is

$$\text{B.t.u. per lb.} = 18{,}600 + 50\,(\text{deg. Bé.} - 10)$$
$$= 13{,}050 + \frac{5{,}600}{\text{specific gravity}}$$

This formula is based upon the specific gravity of the oil and gives results corresponding closely to the calorific values obtained with a calorimeter.

Gravity.—The usual method of indicating the weight of crude or fuel oil is by the Baumé scale. The numbers of this scale are given by the formula:

$$\text{Degrees Baumé} \left\{\frac{140}{\text{s.g.}}\right\} - 130$$

where s.g. is the specific gravity, water being −1−. The specific gravity can be determined by a hydrometer. The gravity of a fuel, in itself, is of no vital importance; however, it can be generally employed as an indication of the oil's freedom from a large residue content and from a heavy percentage of the more complex hydrocarbons that are difficult to burn in a Diesel engine. Table XXVIII is a table of the relation of Baumé gravity to pounds per gallon.

Table XXVIII.—The Baumé Scale

Degrees Baumé	Pounds per U. S. gallon	Specific gravity	Degrees Baumé	Pounds per U. S. gallon	Specific gravity
10	8.336	1.000	26	7.477	0.897
11	8.277	0.993	27	7.435	0.892
12	8.219	0.986	28	7.385	0.886
13	8.161	0.979	29	7.344	0.881
14	8.102	0.972	30	7.294	0.875
15	8.052	0.966	31	7.252	0.870
16	7.994	0.959	32	7.202	0.864
17	7.935	0.952	33	7.160	0.859
18	7.885	0.946	34	7.119	0.854
19	7.835	0.940	35	7.069	0.848
20	7.777	0.933	36	7.027	0.843
21	7.727	0.927	37	6.985	0.838
22	7.677	0.921	38	6.944	0.833
23	7.627	0.915	39	6.902	0.828
24	7.577	0.909	40	6.869	0.824
25	7.527	0.903			

Tank Outage.—Table XXIX gives the amount of oil shortage in tank cars for an oil level a given distance below the tank top.

TABLE XXIX.—OUTAGE TABLE FOR OIL TANK CARS

INCHES OUT	Gallons 6000	Gallons 6648	Gallons 7191	Gallons 7900	Gallons 8050	Gallons 8250	Gallons 10050
1	17	17	16	18	23	20	20
2	47	47	45	50	55	57	56
3	86	88	80	92	101	103	103
4	132	138	120	141	154	159	161
5	183	196	167	197	216	221	226
6	240	259	219	258	283	289	300
7	301	326	276	324	355	363	379
8	366	395	337	394	431	442	461
9	435	468	403	468	512	525	548
10	507	544	474	546	596	612	638
11	583	623	550	628	685	703	730
12	661	705	630	712	775	798	828
13	742	790	712	800	871	895	931
14	825	878	795	890	969	996	1037
15	911	967	881	983	1070	1100	1145
16	998	1059	969	1079	1173	1207	1257
17	1088	1154	1058	1176	1279	1316	1373
18	1180	1250	1149	1276	1386	1427	1491
19	1273	1348	1242	1378	1498	1541	1610
20	1868	1447	1335	1482	1611	1657	1732
21	1465	1549	1429	1588	1726	1774	1857
22	1562	1652	1524	1695	1842	1893	1985
23	1661	1756	1620	1804	1961	2014	2114
24	1762	1862	1717	1914	2080	2137	2244
25	1863	1969	1816	2026	2201	2261	2375
26	1965	2076	1915	2139	2324	2386	2507
27	2068	2186	2015	2252	2448	2513	2642
28	2172	2296	2117	2368	2573	2641	2779
29	2276	2407	2219	2484	2699	2769	2916
30	2381	2519	2323	2601	2827	2899	3055
31	2487	2631	2427	2719	2955	3029	3194
32	2593	2744	2533	2838	3083	3161	3334
33	2700	2858	2639	2957	3212	3292	3476
34	2806	2972	2746	3077	3342	3425	3618
35	2913	3087	2855	3197	3472	3558	3761
36	3020	3203	2964	3318	3603	3691	3906

Dirt in Oil.—Dirt has a deleterious effect on the valves and piston. There is no excuse for any oil being dirty as filtering will remove all the dirt in suspension. While the upper limit of the dirt and coke is set at 10 per cent, a value of half this amount can well be adopted as the upper limit of the residue carried by the oil. For the purpose of removing any dirt, a filter consisting of a flanged tank carrying a brass cylindrical screen which is provided with a felt gasket at the base, the upper flange holding the screen against this gasket has been used in several plants. Terry cloth is wrapped around this screen. The large area of the terry cloth enables the filtering to be carried on at a slow rate. A 1,000 hp. filter, 12 by 24 in. outside diameter, is required to filter $\frac{1}{10}$ gal. per hour per square inch of screen area; this gives a velocity of 0.4 in. per minute. The terry cloth screen may be of as many thicknesses as desired. Three folds ordinarily give ample filtering effect. To clean, the top flange is removed, the screen lifted out and new cloth wound on the screen while the dirty cloth is washed.

A centrifugal filter, such as outlined in Fig. 288, is likewise very efficient as a dirt remover. This type of separator has been largely used in the oil fields to remove the water and dirt out of the oil emulsion peculiar to certain wells.

Flash Point.—The flash point of oil is merely that temperature at which the oil gives off a vapor that will ignite in the presence of an open flame. This point should not be below 250 deg. F. for two reasons. First, if the oil has a lower flash point than that given above, there is a fire risk due to oil vapors forming under atmospheric conditions. Second, if the flash point is low, the oil, as it rests in the engine fuel valve, may vaporize and ignite on account of the absorption of heat from the injection air and the cylinder head. If the flash point is exceedingly high, the ability of the oil to burn in the short time allowed during the admission period is lessened. In this the flash point is merely an indication of the vaporizing characteristics of the oil.

It should be added, at this point, that no person should be allowed to approach an oil storage tank with an open light. Neither should an unprotected electric lamp be dropped into the tank. In cleaning out an empty tank, the cover should be removed and the tank left open for at least forty-eight hours before a man is allowed to enter the tank. The oil vapors dispel very slowly; more than one death is traceable to carelessness in this matter.

Burning Point.—The burning point of an oil is that temperature at which the oil ignites and burns in an open cup. By some it is claimed that the burning point in no way indicates an oil's usefulness as a Diesel fuel.

Nevertheless, if the burning point is high, with a corresponding high flash point, the oil will cause the engine exhaust to be smoky. This is attributable to the sluggish rate of combustion. If the burning point is fairly low, the fuel charge need not be so thoroughly atomized to obtain perfect combustion. As a matter of practical operation, a fuel oil should not have its burning point exceed 300 deg. F. or 150 deg. C.

Sulphur.—The sulphur content should under no circumstance exceed 2 per cent. With this low value the action of the sulphur dioxide on the cast-iron parts is negligible. Even this small amount of sulphur will attack the exhaust pipe line if a water line runs into the exhaust header for cooling purposes. In the presence of the water a formation of sulphuric or sulphurous acid occurs with a consequent eating of the header. On certain early Diesels where water was introduced into the exhaust valve pots, the corrosion from the sulphur was very evident. If the cooling of the cylinder head and exhaust header is carried at a temperature low enough to condense the H_2O resulting from the process of combustion, sulphuric acid may form in the exhaust. This is especially true on low loads when the operator neglects to reduce the flow of cooling water.

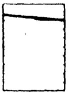
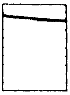
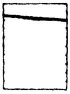
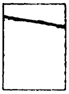
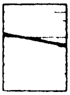
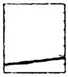
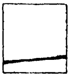
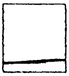
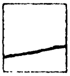
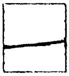
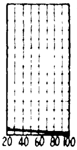
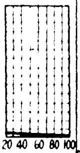
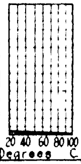
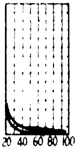
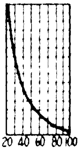
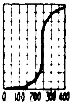
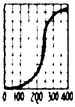
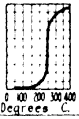
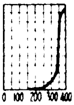
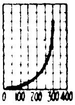
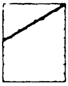
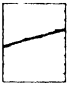
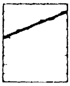
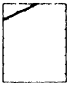
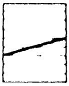
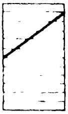
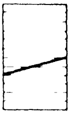
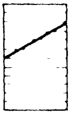
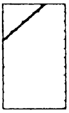
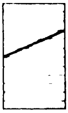
The American Diesels had splash lubrication employing an emulsion of oil and water. The sulphur dioxide gas, blowing past the piston, came in contact with the emulsion, forming sulphuric acid. The acid attacked all the iron and steel parts of the crankcase and cylinder. The same effect may be observed in engines having a water-cooled piston. If the water connections leak, small portions of the joints will show the effects of acid. A corrosive action is at times observed in the cylinder immediately below the head joint. This is undoubtedly the result of water seeping past the flange gasket and uniting with the sulphur in the fuel.

Water.—The presence of water in Diesel fuel oils is objectionable from the viewpoint of both the purchasing agent and the plant operator. It requires no argument to prove that it is of no advantage to buy the water content of the oil. The water increases the net cost of the fuel while likewise increasing the freight charges. Ordinarily this is not of great moment since the percentage of water is never large.

The engineer is justified in protesting against an excessive amount of water since it increases his operating difficulties. In certain fields where the oil comes from flowing wells, instead of from "pumpers," the water and oil exist in an emulsion. The emulsion also entrains a large quantity of air globules. As this combination of oil, water and air enters the fuel pump the air often breaks away from the oil and, collecting in the pump, disturbs the functioning of the valves. The air, water and oil mixture has a greater volume than a pure oil charge of equal heating value. This increased fuel volume demands an increased air-injection pressure to force it into the cylinder.

If the water is in a free state, it will settle at the bottom of the oil storage tank. If this is not drained at frequent intervals, the water level will reach the end of the fuel suction line, with the result that the fuel valve injects nothing but a water charge into the engine cylinder. The engine then drops its load and slows down. This constitutes one of the most serious nuisances

TABLE XXX.—CHARACTERISTICS OF FUEL OILS

			GALICIAN GAS OILS	OKLAHOMA & TEXAS GAS OILS	MEXICAN GAS OILS	TEXAS FUEL OILS	CALIFORNIA CRUDE OILS
1	Lower Heating Value B t u per Pound	19000 18000 17000 16000 15000 14000					
2	Specific Gravity at 15%C	12 11 10 9 8					
3	Viscosity Deg Engler	70 60 50 40 30 20 10 0					
4	Congealing Point	Deg C	+ 5 to - 15	- 1 to - 15	1 to - 10	0 to + 5	0 to - 10
5	Vaporization Curve, Cumulative Per Cent	100 80 60 40 20 0					
6	Flash Point, Deg C	140 120 100 80 60					
7	Burning Point, Deg C	180 160 140 120 100 80 60					
8	Elementary Analysis	C% H% No%	▬ 86 2 ▮ 12 65 ▮ 0 75	▬ 86 2 ▮ 12 6 ▮ 1 0	▬ 84 9 ▮ 12 4 ▮ 0 8	▬ 86 8 ▮ 12 2 ▮ 0 3	▬ 8 ▮ 11.3 ▮ 2 0
9	Ash	Per Cent	None	None	▪ 0 02	▪ 0 02-0 07	0 43 ▬
10	Sulphur	Per Cent	▪ 02-08	▪ 0 2	▬ 1 9	▬ 0 7	▬ 2 5
11	Water	Per Cent	None	None	Trace	Trace	3 2 ▬
12	Free Carbon	Per Cent					
13	Coke Residue	Per Cent	▮ 0 2	▮ 0 3	▮ 0.2	▬ 1-2	6 4 ▬
14	Naphthalene	Per Cent					

that occur in a plant. When it does happen, the entire fuel line must be purged of the water and the storage tank drained. It is practically impossible to secure oil that is entirely free from water. The storage tank can be arranged with a float which will permit the oil suction line to be below the fluid level at all times and yet never touch the bottom where the water rests.

When an engine ceases functioning in all the cylinders, it is usually an indication either of no fuel or of water-logging. If there is no fuel, the engine ordinarily first runs irregularly before stopping. With water entering the cylinder the cessation of ignition is almost instantaneous.

Ash.—The damage resulting from the presence of ash in the fuel is over-estimated. This residue remaining unburnt in the engine cylinder is of a mineral character, being silicate, quartz or iron oxide. There is no doubt that any decided percentage of these substances will score an engine cylinder. Fortunately no oil is offered for Diesel fuel that has an ash percentage above 0.05 per cent.; this amount is so slight that its presence is not revealed through any engine trouble.

Viscosity.—The viscosity of an oil has no bearing on its adaptability as a Diesel fuel. The thickest, most sluggish oil can be made quite free-flowing through the application of heat. In the Northern states every oil storage tank should have heating coils for use in the winter months. If these coils are connected with the discharge cooling water before it enters the cooling tower, the heat abstracted is ample to make the oil free-flowing. In the cold climates it is always advisable to have a tank inside the building of at least one barrel capacity. If shut-downs are of more than a few hours duration, the engine should be run on kerosene or stove oil until the heavy oil becomes fluid. It is necessary to drain the fuel pump and lines on a shut-down to prevent clogging of the lines with congealed oil.

Acid.—To avoid corrosion in the pumping and cylinder parts the oil should be devoid of any trace of acid.

Typical Oils.—In Table XXX are shown the characteristics of a number of oils. The Baumé gravity may be found by comparing the specific gravities with those in Table XXVIII.

Navy Oil Specifications.—The following cover the specifications of the United States Navy. It will be noticed that the oil for the submarine Diesels is of better grade than the standard Diesel fuel.

Special Fuel Oil for Diesel Engines in Submarines.—The specification covers the grade of oil used by the United States Government and its agencies as a fuel for Diesel engines in submarines.

Fuel oil shall be a hydrocarbon oil, free from grit, acid, and fibrous or other foreign matters likely to clog or injure the burners or valves. If required, it shall be strained by being drawn through filters of wire gauze of 16 meshes to the inch. The clearance through the strainer shall be at least twice the area of suction the pipe, and the strainers shall be in duplicate.

Properties and Tests

Flash Point.—Method 110.21. The flash point shall not be lower than 150 deg. F. (Pensky-Martens closed tester).

Water and Sediment.—Method 300.3. Water and sediment combined shall not amount to more than 0.2 per cent.

Carbon Residue.—Method 500.1. The carbon residue shall not exceed 0.5 per cent.

Precipitation Number.—Method 310.1. The precipitation number shall not be greater than 0.5.

All tests shall be made according to the methods for testing petroleum products contained in Part 2 of Technical paper No. 323, Bureau of Mines.

Fuel Oil (Navy Standard).—This specification covers the grade of oil used by the United States Government and its agencies where a high-grade fuel oil is required. This oil may be used in Diesel engines.

Fuel oil shall be a hydrocarbon oil, free fromgr it, acid, and fibrous or other foreign matters likely to clog or injure the burners or valves. If required, it shall be strained by being drawn through filters of wire gauze of 16 meshes to the inch. The clearance through the strainer shall be at least twice the area of the suction pipe, and the strainers shall be in duplicate.

Properties and Tests

Flash Point.—Method 110.21. The flash point shall not be lower than 150 deg. F. (Pensky-Martens closed tester). In case of oils having viscosity greater than 30 seconds at 150 deg. F. (Saybolt Furol viscosimeter) (8 deg. Engler), the flash point shall not be below the temperature at which the oil has a viscosity of 30 seconds.

Viscosity.—Method 30.3. The viscosity shall not be greater than 100 seconds at 77 deg. F. (Saybolt Furol viscosimeter).

Sulphur.—Method 520.21. Sulphur shall not be over 1.5 per cent.

Water and Sediment.—Method 300.3. Water and sediment combined shall not amount to over 1 per cent.

All tests shall be made according to the methods for testing petroleum products contained in Part 2 of this paper.

Bunker Fuel Oil "A."—This specification covers the grade of fuel oil used by the United States Government and its agencies where a low-viscosity oil is required. This oil may be used in Diesel engines.

Fuel oil shall be a hydrocarbon oil, free from grit, acid, and fibrous or other foreign matters likely to clog or injure the burners or valves. If required, it shall be strained by being drawn through filters of wire gauze of 16 meshes to the inch. The clearance through the strainer shall be at least twice the area of the suction pipe, and the strainers shall be in duplicate.

PROPERTIES AND TESTS

Flash Point.—Method 110.21. The flash point shall be not lower than 150 deg. F. (Pensky-Martens closed tester). In case of oils having viscosity greater than 30 seconds at 150 deg. F. (Saybolt Furol viscosimeter) (8 deg. Engler), the flash point shall not be below the temperature at which the oil has a viscosity of 30 seconds.

Viscosity.—Method 30.3. The viscosity shall not be greater than 100 seconds at 77 deg. F. (Saybolt Furol viscosimeter).

Water and Sediment.—Method 300.3. Water and sediment combined shall not amount to over 1.0 per cent.

All tests shall be made according to the methods for testing petroleum products contained in Part 2 of paper No. 323.

Bunker Fuel Oil "B."—This specification covers the grade of fuel oil used by the United States Government and its agencies where a more viscous oil than bunker oil "A" can be used. This oil may be used in Diesel engines of a type adapted to an oil of medium viscosity.

Fuel oil shall be a hydrocarbon oil, free from grit, acid, and fibrous or other foreign matters likely to clog or injure the burners or valves. If required, it shall be strained by being drawn through filters or wire gauze of 16 meshes to the inch. The clearance through the strainer shall be at least twice the area of the suction pipe, and the strainers shall be in duplicate.

PROPERTIES AND TESTS

Flash Point.—Method 110.21. The flash point shall be not lower than 150 deg. F. (Pensky-Martens closed tester).

Viscosity.—Method 30.2. The viscosity shall not be greater than 100 seconds at 122 deg. F. (Saybolt Furol viscosimeter).

Water and Sediment.—Method 300.3. Water and sediment combined shall not amount to over 1.0 per cent.

All tests shall be made according to the methods for testing petroleum products contained in Part 2 of paper No. 323.

Bunker Fuel Oil "C."—This specification covers the grade of fuel oil used by the United States Government and its agencies where a high-viscosity oil is satisfactory. This oil may be used in Diesel engines of a type adapted to an oil of high viscosity.

Fuel oil shall be a hydrocarbon oil, free from grit, acid, and fibrous or other foreign matters likely to clog or injure the burners or valves. If required, it shall be strained by being drawn through filters of wire gauze of 16 meshes to the inch. The clearance through the strainer shall be at least twice the area of the suction pipe, and the strainers shall be in duplicate.

PROPERTIES AND TESTS

Flash Point.—Method 110.21. The flash point shall be not lower than 150 deg. F. (Pensky-Martens closed tester).

Viscosity.—The viscosity shall not be greater than 300 seconds at 122 deg. F. (Saybolt Furol viscosimeter).

Water.—Method 300.1. The water content shall not be more than 1.0 per cent.

NOTE—Oil may be accepted containing not more than 2.0 per cent of water. In this case a deduction in price for all water over 1.0 per cent shall be made.

Sediment.—Method 300.2. The sediment content shall not be more than 0.25 per cent.

All tests shall be made according to the methods for testing petroleum products contained in Part 2 of this paper.

Engine Oil Tanks.—Each engine should be supplied with an individual oil tank. This tank is best elevated with its base at least 5 ft. above the engine fuel pump. This places a pressure head on the pump suction that does much to eliminates air leakage into the pump. The tanks are often of two compartments, one of which holds kerosene for starting purposes. The employment of kerosene for starting is advisable in all plants. It reduces the liability of the engine refusing to fire and eliminates the tarry deposits that frequently occur when the heavy fuel oil is injected into a cold engine. The kerosene should be fed for 10 or 15 minutes on starting and for 4 or 5 minutes before stopping. At the latter time,the kerosene cuts the carbon deposits and prevents freezing of the piston rings.

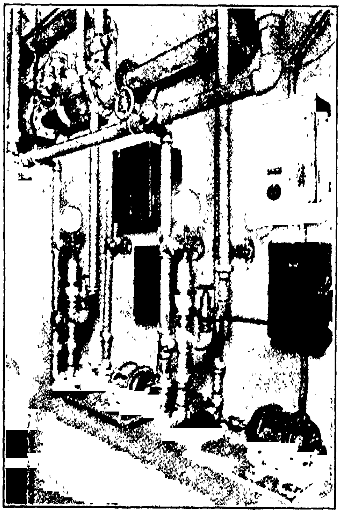
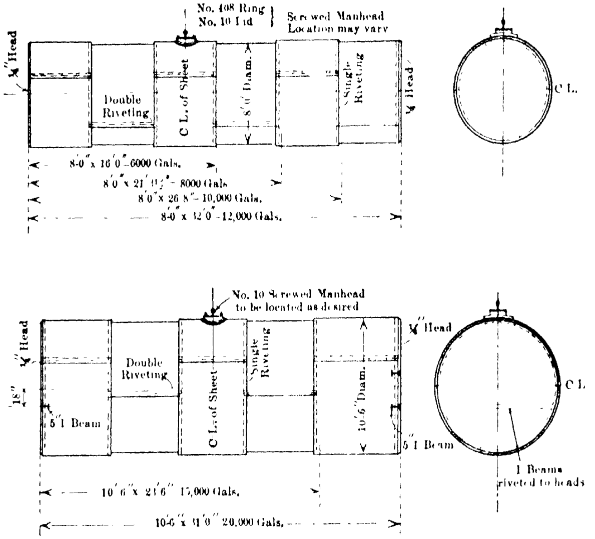


FIG. 292.—Fuel-oil pumps of a 3,000 hp. plant.

The tank, in every instance, ought to have a level indicator fitted to it. A simple indicator can be made out of a galvanized float, two pulleys and a light-weighted pointer. The oil can be pumped into this tank from the storage tank by either a hand or motor-driven pump. Figure 292 illustrates a typical fuel pumping installation for a 3,000-hp. Diesel plant. For a plant of less than 300 hp. the quantity of oil handled does not justify the expense of a motor-driven pump; a hand rotary pump proves quite suitable and is low in cost.

Oil Storage Tanks.—The capacity of the plant together with its location largely determines the type of oil storage tank. If the plant consumes less than 12,000 gal. per year, two steel tanks of 6,000 gal. each are probably the most advisable size to install. The employment of two tanks provides means whereby a damaged tank can be repaired without a plant shut-down. It allows a thorough cleaning .of the tank after each emptying. Figure 293 covers the steel tanks that are of commercial sizes.



Size	Capacity	Heads	Shell	Riveted with	Weight
8′ × 16′	6,000 Gals.	¼″	3/16″	½″ Rivets	5,000 lbs.
8′ × 21′-3½″	8,000 Gals.	¼″	3/16″	½″ Rivets	6,100 lbs.
8′ × 26′-8″	10,000 Gals.	¼″	3/16′	½″ Rivets	7,200 lbs.
8′ × 32′	12,000 Gals	¼″	3/16″	½″ Rivets	8,400 lbs.
10′-6″ × 23′-6″	15,000 Gals	¼″	3/16″	½″ Rivets	9,000 lbs
10′-6″ × 31′	20,000 Gals.	¼″	3/16″	½″ Rivets	11,000 lbs.
8′ × 16′	6,000 Gals.	¼″	¼″	½″ Rivets	6,100 lbs.
8′ × 21′-3½″	8,000 Gals.	¼″	¼″	½″ Rivets	7,600 lbs.
8′ × 26′-8″	10,000 Gals.	¼″	¼″	½″ Rivets	9,100 lbs.
8′ × 32′	12,000 Gals.	¼″	¼″	½″ Rivets	10,600 lbs.
10′-6″ × 23′-6″	15,000 Gals.	¼″	¼″	½″ Rivets	11,000 lbs
10′-6″ × 31′	20,000 Gals.	¼″	¼″	½″ Rivets	14,000 lbs.

A 2″ Duplex Oil Tank Vacuum Valve should be furnished with each tank.

Fig. 293.—Steel oil storage tank.

If steel tanks are installed, they should be set in a pit and covered to eliminate any fire risk. Figure 294 illustrates a convenient method of placing the tanks. Concrete saddles are placed in the pit; the tanks rest on these saddles. The earth is filled in and a 2- to 4-in. layer of concrete is formed around the tank, completely encasing it. This concrete layer prevents any corrosion of the steel plate by excluding all air and moisture.

Where the ground is inclined to be marshy or water-logged at periods of wet weather, an empty steel tank has sufficient buoyancy to float on this water and, rising up above the ground, will break the pipe connections. To avoid this danger, in ground of this character, a heavy concrete slab can be run about the upper half of the tank or concrete beams laid across the top of the tanks and tied to the saddles with long rods. The weight of the concrete then overcomes the buoyancy of the tank.

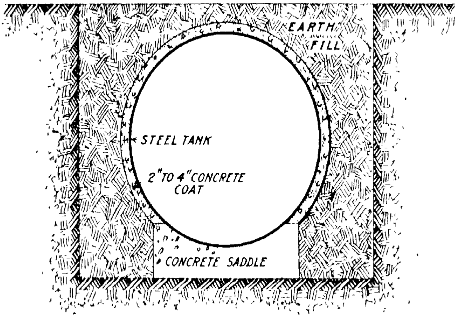


FIG. 294.—Steel tank setting.

Concrete Storage Tanks.—From every viewpoint the concrete storage tank is the most advisable to install. It is cheaper than a steel tank of a corresponding capacity and has an indefinite life with no danger of leakage if the concrete is well reinforced.

Figure 295 sketches a circular concrete tank that can be made of any desired capacity. The bottom must be well reinforced with steel rods or hog wire fencing. The vertical walls should have 5/8- to 3/4-in. iron rods every 8 to 12 in. To form the top, concrete beams can be placed on the vertical walls and covered with concrete slabs; these should be reinforced with wire fencing.

Coal Tar Oils.—Coal tar, as a Diesel fuel, has received no attention in the United States. This is due to the low price of the petroleum products. While it will be years before the American Diesel operator will be confronted with the problem of coal tar as fuel, this fuel has its future possibilities. There are several American Diesel engines operating on Mexican crude that can be easily adapted to tar oils, providing a fairly constant load is carried. Under such conditions the atomizer will nebulize the tar oil sufficiently for ignition with the employment of a primary ignition oil.

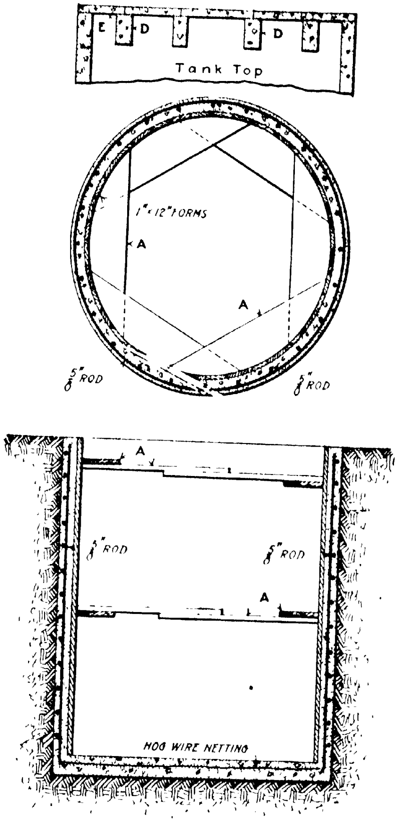


Fig. 295.—Concrete oil storage tank.

Methods of Burning Tar Oils.—A number of methods have been devised to burn the tar oil successfully. The following list comprises those that have given fairly satisfactory results.

1. Mixture of tar oil and petroleum oil.
2. Tar oil for full load and petroleum for light loads.
3. Raising thermal range of the engine cycle.
4. Increasing the compression.
5. Application of a light oil for primary ignition.
6. Catalytical action.

1. *Mixture of Tar Oil and Petroleum Oil.*—Among the first plans advocated was a mixture of tar oil and a lighter petroleum oil. Various experiments have been made with a varying proportion of coal tar. With a constant load the coal tar percentage can be as high as 75 per cent without affecting the functioning of the engine. The problem that so far has not been solved is the thorough mixing of the two oils. If a mechanical agitator is placed in the engine fuel tank, a separation of the oils will occur in their passage to the fuel valve. The consequence is an irregularity in the ignition of the charges. This method can hardly be termed successful at the present time.

2. *Tar Oil for Full Load and Petroleum Oil for Light Loads.*—There is no condition existing within the engine cylinder that would prevent this method from being successful. However, the design of the fuel injection system presents many mechanical difficulties. The injection valve as well as the fuel pump must be under governor control.

3. *Raising Thermal Range of the Engine Cycle.*—This method includes the raising of the temperature of the injection air, the fuel charge and the cylinder air charge; in fact, it contemplates the increase of the temperature of the entire working parts. Starting with a higher temperature of the cylinder charge of air, the combustion of the injected fuel takes place at an increased temperature. This insures a positive ignition of the tar oil. The thermal efficiency of the engine depends on the temperature range $t_1 - t_2$. Although the actual temperatures t_1 and t_2 are higher than usual, the range $t_1 - t_2$ is identical with the normal Diesel temperature range. Consequently the efficiency would not be changed. The objection lies in the increased stresses occurring in the engine, and the method has never met with commerical success.

4. *Increasing the Compression.*—The plan of increasing the compression has received considerable attention but is not commercially attractive. The Diesel engine under present designs works with as great a pressure as is advisable. The increase

of compression pressure would augment the operating difficulties and is dangerous. Furthermore, many tests of engines with varying compressions appear to bear out the belief of practical engineers that a compression of 500 to 550 lb. per square inch gives the engine the greatest possible efficiency and is as high as is practical. The injection-air pressure would necessarily be increased a corresponding amount, thus increasing the engine losses. In these tests referred to, the engine smoked badly at loads below half rating.

5. *Application of a Light Oil for Primary Ignition.*—This plan is the only one offered that possesses merit and has been employed to a large extent in Germany and in a few English installations. The fuel-injection valve, which, in most cases, is of the open type, is equipped with two oil lines. All of the tar-oil valves are designed to allow the light ignition oil to be blown into the cylinder ahead of the tar-oil charge. Being lighter, this primary charge immediately ignites and supplies the additional heat required to ignite the heavy tar oil. The time interval for this action is short even in a slow-speed engine. Consequently, it cannot be expected that an engine above 200 r.p.m. will successfully handle tar oils even under this system. The fuel valve becomes foul in a short time. A thorough cleaning is necessary at least once a week. The tar coats the piston and combustion-chamber walls with a thick hard scale that requires constant attention.

6. *Catalytical Action.*—No commercial engine has been operated under this method of tar-oil ignition. A catalytic agent is required that will produce combustion even when the tar is in fairly large particles. The cost of the agent and its life are matters of importance. It is highly improbable that it will ever be commercially successful.

Use of Vegetable Oils.—In the United States by reason of the supply of petroleum there is little interest shown in the use of vegetable oil. However, in the Tropics fuel oil is high in price while native stocks of vegetable oils are frequently cheap. For this reason the adaptability of oil engines for using these oils is of importance. The accompanying table gives the results obtained by M. Mathot on European engines of the semi-Diesel type using palm oil. The increased fuel consumption when using the vegetable oil is not great in case of any of the engines.

TABLE XXXI.—TESTS WITH VEGETABLE OILS

Make and type (bore and stroke), in	Date of test	B hp	R p m	M e p, lb per sq in	Oil consumption per B hp-hr in pounds		
					Palm	Cotton	Mineral heavy
1. Gardner, Patricroft, two-cycle vertical ($8\frac{1}{2} \times 9\frac{1}{2}$)	1920 June	16.25	356	41.0	0.71	0.71	0.56
		22.0				0.57	0.52
2. Gardner, four-cycle ($9\frac{1}{2} \times 17$)	June	25.0	228.32	82.0		0.59	
		12.0				0.65	0.60
3. Kromhout, Amsterdam, two-cycle ($11\frac{13}{16} \times 12\frac{1}{4}$)	Oct	33.5	335	37.0	0.64		
		35.7	333	38.4		0.61	
		23.5	406	41.0			0.57
		27.3	411				0.64
4. Ansaldo, Turin, two-cycle ($9\frac{1}{2} \times 16\frac{7}{8}$)	1921 Feb	21.1	423		0.69		
		24.0			0.72		
		22.8	412			0.66	
		25.7	413			0.75	
5. Anglo-Belgian, Belgium, two-cycle ($10\frac{1}{4} \times 11\frac{13}{16}$)	1921 March	25.0	384	33.0			0.54
		25.5			0.69		

CHAPTER XIX

LUBRICATION

LUBRICATION SYSTEMS. LUBRICATION SPECIFICATIONS. OPERATING DIFFICULTIES

General.—The problem of lubrication is one of the most important that confronts the Diesel engineer. Some individual engines have achieved a sorry reputation due to poor lubrication facilities or to mediocre grades of lubricating oils. The engineer does well to insist on a high quality of oil, and he should see that the oiling devices function properly. If the lubrication method is incorrect, a proper device should be installed.

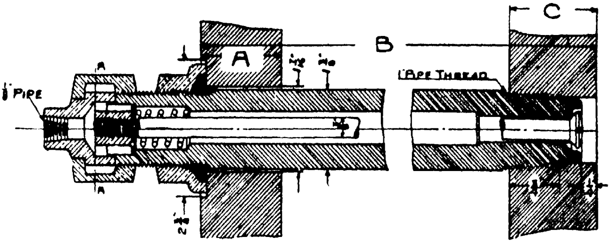
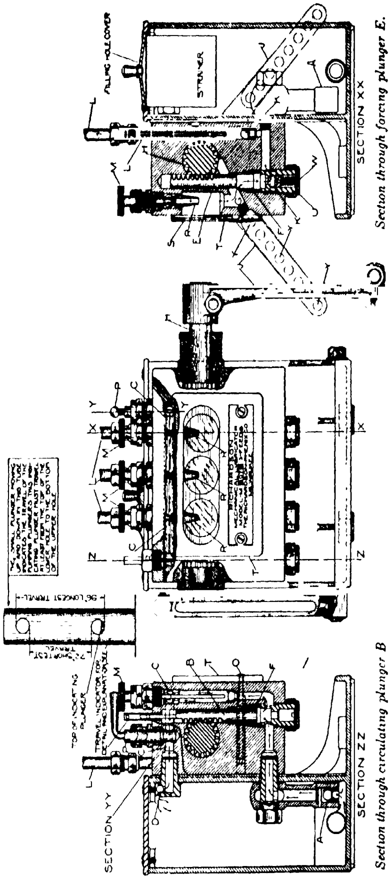


FIG. 296.—Cylinder lubrication check valve.

Cylinder Lubrication Systems.—The most popular cylinder lubrication system is the force-feed design where one or more positive-driven mechanical oil pumps force the lubricant to all moving parts with the exception of the crank bearings. These parts are generally oiled by gravity stream lubrication. A design of cylinder oiling that is coming into use employs a spray check valve quite similar to the fuel injection nozzle of a low-pressure engine. This valve, Fig. 296, passes through the cylinder jacket and cylinder wall. The mechanical oil pump is timed to cause the lubricating oil to be injected through this nozzle when the piston is at outer, or crank, dead-center, the oil entering below the top ring. The oil sprays on to the piston, covering a considerable area even though the clearance between the piston and cylinder walls is small. The piston, as it moves upward, swabs this oil over the cylinder walls. This principle of



Section through forcing plunger E.

Section through circulating plunger B

Fig. 297.—Mechanical lubrication pump used on Diesel engines.

cylinder lubrication is most satisfactory in operation. The oil is deposited when the cylinder temperature is fairly low; consequently a decided lubricating effect will be experienced before the flame of combustion burns the lubricating oil.

Even when such an injection device is not employed, the oil is always forced into the cylinder by a mechanical oil pump. Although several makes of oil pumps are in use, the majority of domestic Diesels are equipped with the mechanical oil pump shown in Fig. 297, where the course of the oil is indicated by arrows.

An engineer should bear in mind that the mechanical oil pump is subjected to stoppage of one or more oil lines due to the presence of bits of waste. Consequently, as much attention must be given to the pump as to other parts of the engine.

It is of the greatest importance that, having installed a mechanical lubricator, the operator should be certain that the lubricator employed is capable of delivering the oil at the proper time and in the correct amount for long periods of time and over a wide range of load variation and temperatures. It is also essential that the lubricator should be capable of a wide range of adjustment enabling the feeds to be cut down to a very low point without stopping the flow. There are many designs of mechanical lubricators which, although very compact, leave much to be desired with regard to accessibility of the various parts.

Some forms of sight feeds show oil that is *not* going to the engine, whereas what is wanted is to see the actual oil and the amount that is actually being fed. It is advisable also to have the oil flow timed and to know the moment at which the oil is being fed in relation to the position of the piston or other moving part to be lubricated at that moment.

The best type of machine is undoubtedly a pump which depends entirely for the amount of oil discharged on the displacement of the ram, the pump chamber being always completely flooded. The action of the lubricator is, with design, independent of the viscosity of the oil and of the head level of the oil in the tank, and being independent of viscosity, it is also independent of changes in temperature within wide limits. An efficient strainer should always be supplied regardless of the type of lubricator.

Cylinder Lubrication.—Considering first the lubrication of the power cylinder, the lubricating oil must do two things—lubricate the cylinder walls, piston and piston rings, and assist the piston rings to form a seal. As the greater proportion of power-plant engines operate on the four-stroke cycle, let us first investigate what happens to the oil on each of these four strokes.

On the charging stroke the cylinder walls, as far as the piston travels, have been covered with a film of oil from the previous stroke, which, together with the oil on the piston and between the rings, acts as a lubricant for the piston and rings. The oil in the clearance space between the rings assists them in forming a seal to prevent leakage of air or vapor from the crankcase into the cylinder. The air and vapor drawn into the cylinder is comparatively cool, and if the fuel is completely vaporized, there will be no detrimental effect on the film.

The film of oil left on the cylinder walls during the suction stroke, together with the new film picked up by the piston from the cylinder walls, lubricates the piston on the compression stroke, and as the piston moves toward the cylinder head it smears a fresh film of oil on the walls. The supply of oil collecting on the advancing side of the piston rings and around the upper edge of the piston serves to prevent leakage of the gases being compressed. The degree of compression will depend, therefore, somewhat upon the seal-forming properties of the oil and on the condition of the cylinder, piston and rings.

During the compression stroke the temperature of the gases gradually increases, but does not rise high enough to do any damage to the oil film until near the end of the stroke, when most of the oil film is covered by the piston.

The film of oil placed on the cylinder walls by the piston on the compression stroke lubricates the piston on the working stroke. As the piston moves toward the crank, exposing the walls to the high temperature of the burning gases, the flame comes in contact with the oil film, but only after it has served its purpose of lubricating the piston on the in-stroke. The greater part of the damage to the oil film occurs on this stroke. The oil between the rings and between the piston and cylinder walls is subjected to the pressure of the burning gases and must assist the piston rings in preventing loss of power through leakage of the gases out of the open end of the cylinder.

If there is any trouble with lubrication, it will be encountered on the exhaust stroke, as the oil film on the cylinder walls has just been exposed to the high temperatures of the burning gases and has been more or less damaged. However, if the oil possesses the proper characteristics and has been applied in sufficient amount, some lubricating value remains, which, together with the oil film on the piston itself, lubricates the piston for this stroke.

Although it is a difficult matter to make accurate determinations of the actual temperatures within the cylinders, estimates made by various authorities are close enough for our purpose. The maximum temperature obtained immediately after firing is about 2,700 deg. F. and the minimum temperature during suction stroke, 250 deg. F. The average temperature for complete cycle is 950 deg. F. It should be remembered that these are temperatures of the gases and not the temperatures of the cylinder walls or of the oil film.

The inner surface of the cylinder wall probably ranges from 30 to 60 deg. F. hotter than the cooling water, depending on the thickness of the metal and the efficiency of the cooling system. As long as the water is not boiling, the cylinder-wall temperatures will hardly be more than 260 to 270 deg. F., and with a normal cooling-water temperature of 110 to 120 deg. F., at which many power-plant engines operate, the cylinder-wall temperature is below 175 deg. F. The oil film itself is hotter than this, at least on that side of the film exposed to the burning gases. The maximum temperature of 2,700 deg. F. occurs only when the piston moves toward the crank, uncovering the oil film and exposing it to the flame until near the end of the stroke. The inner surface of the oil film in contact with the cylinder wall at the comparatively low temperature of 175 to 200 deg. F. is affected very little by the high temperatures, but the outer surface of the oil film directly exposed to the flame is undoubtedly damaged.

As it is impossible to produce a petroleum lubricating oil of any kind having a flash point over 700 deg. F., it would seem that the oil film exposed to an average temperature of 950 deg. F. would be destroyed promptly. That such is not the case the engineer can readily prove to himself by pouring a few drops of oil on a red-hot plate. It will be several seconds before the oil is completely evaporated. If the engine is running 200 r.p.m., the oil film will be replaced three times a second. So it is a comparatively simple matter to maintain the oil film if oils of suitable characteristics are used.

Influence of Completeness and Rapidity of Combustion.—The completeness of combustion has a great influence on the kind of lubricant to be used and its life. If combustion is completed very soon after the fuel-spray valve closes, maximum temperatures are obtained when the piston covers the working surfaces of the cylinder, thus protecting the lubricating film from destruction. If the fuel is not completely vaporized and intimately mixed with air in the right proportions, it burns slowly, and as the piston uncovers the oil film, the burning fuel destroys it. An excessively lean mixture, such as exists on low loads by reason of the small amount of fuel injected into the cylinder will allow, by reason of the large excess of air, the oil film to be completely burned up or at least so badly coked as to form carbon deposits. The bad effects of slow burning can be overcome sometimes by using a heavy-bodied oil.

Piston Seal.—When there is a film of oil on the cylinder, if it is of the right thickness and the oil possesses the proper characteristics, the friction of a tight-fitting ring can be lessened and the seal of the loose-fitting ring improved. Taking as an example, the vertical cylinder of a four-stroke-cycle engine, as the piston moves downward on the suction stroke, the lower edge of the ring scrapes a portion of the oil off the cylinder wall, forming a considerable body of oil on the advancing side of the ring. This oil "wave" reduces the passage of air and oil vapor from the crankcase into the cylinder past the rings, and allows the piston to draw in a full charge of air.

On the compression stroke the oil builds up on the top edge of the ring, and so prevents the blowing of the compressed vapor past the rings. As the piston is forced down on the working stroke, the pressure is exerted upon the oil resting on the top side of the ring. If there is only one ring, most of this oil would probably be blown out of the piston clearance space by the high-pressure gases, but when there are two or more rings, any oil blown away from the top ring, packs up against the second ring and tends to prevent the further passage of the gases.

Another place for leakage is through the clearance space back of the rings, between the ring and its groove. If the ring is a loose fit in the groove particularly with respect to width, the vapor will find its way around the ring and power will be lost. The oil is of benefit here too, as it fills this clearance space, and its resistance helps to prevent the gas leakage. The value of the oil in assisting the rings is particularly noticeable where the cylinder is not perfectly round, so that it is impossible for the rings to prevent leakage of gases without the assistance of the oil.

Effect of Viscosity.—The physical properties of the oil that influence its seal-forming value are viscosity, surface tension with respect to metal, and cohesion. There may also be some chemical reaction between certain constituents of some kinds of oils and the metal of the cylinder and the piston which increases its adhesion to the metal and prevents the oil from being scraped off.

Other things being equal, one might expect that a high viscosity would cause the oil to offer a greater resistance to being blown out of the clearance space and would also permit the accumulation of a larger body of oil on the advancing side of the rings than could be secured by a lower-viscosity oil. It is found by actual experience that the piston seal can be greatly improved in many instances by raising the viscosity of the oil; better compression is secured and more power developed by the engine at a lower consumption of fuel per horsepower. This is especially true where the piston, cylinder and rings have become worn.

From this statement the conclusion may be drawn that if a high-viscosity oil is good, a higher viscosity would be better. This is not true, as it is sometimes found that if the viscosity is too high the oil will not work its way between the piston and the cylinder fast enough, the cylinder becomes dry, and many troubles follow the lack of sufficient oil.

Just what the viscosity should be depends upon the design of the engine. In general a viscosity of from 500 to 700 seconds, Saybolt Viscosimeter, at 100 deg. F. covers the oils ordinarily used.

Should the viscosity be a little low, there is not likely to be lack of lubrication, but simply loss of power through leakage and troubles from excessive oil in the cylinder and carbon deposits which gum the piston rings. The oil must be of just the right viscosity, and since the viscosity varies with the temperature, it must have the right viscosity at the working temperature of the oil film. Some engines run hotter than others and need a heavy oil. Others run cool and a viscous oil could not feed fast enough to give proper lubrication.

An oil with a low surface tension with respect to the metals of which the cylinder and piston are composed, will adhere more tightly to them than an oil with a higher surface tension; in other words, its adhesiveness will be greater. A high adhesive quality will also prevent the oil from being blown off the cylinder wall and out of the clearance space and therefore will improve the piston seal. Unfortunately, it is a difficult matter at present to measure the adhesion of oils with respect to different metals, and to depend entirely upon actual running tests to determine the value of the various lubricants.

Stream Lubrication.—As has been mentioned, an engine having a mechanical oiler for the cylinders, as a rule, employs either drop, stream, or pressure lubrication for the bearings. The stream system is preferred by some builders since a sufficient amount of oil is assured at all times.

The lubrication of the crankpin is usually accomplished, where stream lubrication is used, by fitting banjo oil rings to one of the throws of each crank. A drilled passage in the crankpin leads the oil from the banjo to the crankpin bearing.

The piston pin, with such a system, is oiled in one of a number of ways. Some engines have the piston pin lubricated through a groove cut in the side of the piston with a drilled passage leading to the pin. Oil from a timed mechanical lubricator is forced onto the cylinder wall at the time that this groove registers with the oil line nozzle.

Pressure-feed Lubrication System.—A pressure-feed system for crankshaft and connecting-rod bearings is employed on the Busch-Sulzer Type B, and most other Diesels. This system, as used on the Busch-Sulzer, is outlined in Fig. 298, showing the drilled crankshaft and connecting rods, as well as the pump and filtering mechanism. The oil is forced by the pump, which is a rotary geared to the air-compressor crank disk, into the lower half of the main bearings as indicated. The oil flows around the journal, lubricating the entire surface. The diagonal-drilled passages to the crank pins once each revolution register with the oil-inlet opening in the lower bearing shell. This allows a stream of oil to enter the passage and to flow to the crank-pin box. The connecting rod also is drilled, and each revolution this passage aligns with the crank end of the diagonal passage, which causes a supply of oil to pass up the rod to the wrist-pin bearing. The drip from the various parts is caught in the crankcase and from there flows through the filter into the cooler. The pump then draws the oil from the cooler to be reused. In order to seal the various parts of the system, the oil level in the crankcase should be around 3 in. before the engine is started and should always be carried at least 1 in. in depth while the engine is running. The pump discharge has a pressure gage attached; the pressure should average around 25 lb. The gage pressure is a fair indication of the condition of the system. If the pressure shows a decided increase, it is probable that part of the discharge lines are clogged with dirt or waste.

The presence of water in the lubricating oil will cause the pressure to rise. Where the water is fresh or pure, the oil containing it should be drawn off and the water settled out by heating. In regard to sea water getting into the lubrication system, every marine engineer avoids this as he would the plague,

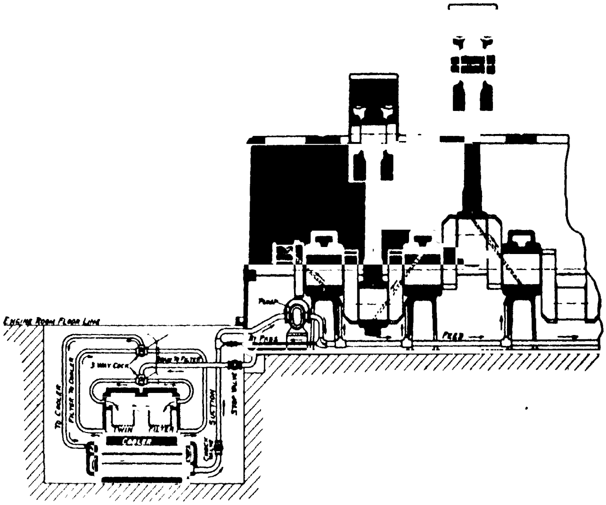


FIG. 298. Busch-Sulzer lubrication system.

and it amounts to a state of affairs so serious that it is frequently the cheapest to take no risks, and entirely replace the system with new oil. Sometimes annoying leaks occur from the water-cooled exhaust valves or cages, and the water finds its way into the oil system. Trouble also arises from leakage of water at the point where the oil passes to the pistons through the jacket, although such cases are not frequent. Where the pistons also are water-cooled a close look-out should be kept for leakage of water, and where the pistons are oil-cooled internally, the oil selected should be of high quality. Care should also be exercised when dismantling the valves, cylinder covers, and when the water connections are broken, to ensure that water is excluded from the lubrication system.

Should there be a considerable fall of pressure, as shown by the gauge, frequent examinations of the lubricants from the filters should be made, and if it is certain the pumps are in order, make sure by observation and smell that no fuel oil has found its way to the crankcase, or has passed the pistons unburnt.

Worn bearings will, likewise, cause a drop in pressure due to the escape of oil at the bearing ends.

Where the pistons are water-cooled, at times water mixes with the oil in the crankcase. If in any considerable amount, there is danger of bearing trouble. A water separator, such as the De Laval, will remove any entrained water. It should be connected between the crankcase and the oil filter.

Splash Oiling Systems.—The splash oiling system is almost abandoned in Diesel-engine practice. It was, however, the method employed on the first American Diesels for lubricating the bearings and connecting-rod brasses as well as the piston. The American Diesel Co. used a mechanical oil pump for the purpose of lubricating the cylinder walls but, in fact, depended on the splash from the crankcase to meet most of the cylinder lubrication demands.

The great objection to the splash system is the difficulty of preventing excessive oil deposits forming on the cylinder walls;

Cylinder Lubrication Oil Specifications.—While oils of widely varying characteristics give excellent service in many plants, the following specifications, if adhered to, will guarantee the procurement of an oil that will give satisfactory results. Practically all refineries have a lubricant that will meet these requirements.

LUBRICATING OIL SPECIFICATIONS

	DEG. F.
Boiling point	600–700
Flash point	325–500
Fire point	400–600
Viscosity at 100 deg. F., Sayboldt	550–800
Specific gravity, Baumé	18–24
Carbon content, per cent	0 05–0 2
Sulphur	None

If the lubricating oil is to be used on the compressor cylinders, the flash point must be above 450 deg.

Bearing Lubrication.—The viscosity of the oil used on the bearing may vary over a wide range and still be satisfactory. The oil with the lower viscosity, all other things being equal, will cause the bearings to run at a lower temperature but even with a high viscosity oil after a time the bearing temperature becomes constant.

Testing Lubricating Oils.—The method to be followed in testing internal-combustion cylinder oils depends a good deal upon what sort of information is desired. To compare the relative suitability of two oils for a power-plant engine, we must test them under identical conditions. The average operating load and speed should be selected, or if the load is variable, the test should be made long enough to average up the loads on both tests. Although it will not be feasible in some instances to secure all the information suggested, the following log recommended by W. F. Osborne of the Texas Co. gives a list of the data usually needed when comparing oils.

A. TIME:
  1. Time of starting the test.
  2. Time of each reading.
  3. Time of ending the test.
  4. Total number of hours in operation.

B. OIL CONSUMPTION:
  1. Gallons of oil put into lubricating system or lubricator at start.
  2. Gallons of oil added to system or lubricator during test
  3. Gallons in system or lubricator at end of test.
  4. Calculate gallons consumed per brake horsepower per hour or per kilowatt-hour.

C. FUEL CONSUMPTION:
  1. Fuel in tank or reservoir at start of test.
  2. Fuel added during test (use meters for gas).
  3. Fuel in tank or reservoir at end of test.
  4. Calculate pounds of liquid fuel or cubic feet of gas brake horsepower per hour or per kilowatt-hour output.

D. LOAD CONDITIONS:
  1. Revolutions per minute.
  2. Brake horsepower by prony brake or dynamometer at ¼, ½, ¾ and full load.
  3. Horsepower by indicator diagrams at loads as in D-2 and also for no load.
  4. Calculate mechanical efficiency at various loads and plot curve.
  5. Calculate frictional losses at various loads and plot curve.

E. INFLUENCING OPERATING CONDITIONS:
  1. Temperature of engine room.
  2. Temperature of circulating water to engine

3. Temperature of circulating water from engine.
4. Gallons of water circulated per hour.
5. Temperature of oil in crankcase (if crankcase oiling is used).
6. Temperature of oil to bearings in case of mechanical lubricator.
7. Temperature of air to cylinder.
8. Temperature of exhaust gas.
9. Time of ignition.
10. Analysis of samples of exhaust gas to determine efficiency of combustion.
11. Analysis of samples of fuel to determine constancy of fuel.

F. CONDITION OF ENGINE AT END OF TEST:

1. Condition of cylinder head, piston, rings and valves, with respect to wear, polish and carbon deposits.
2. Character and amount of carbon deposits.
3. Analysis of crankcase oils to show deterioration of lubricant. (Samples should be taken at intervals during the actual running of the test.)
4. Analysis of new oil for all physical and chemical properties.

If a test is made to compare the lubricating performance of one oil on two or more engines, in addition to the data herein indicated, complete information concerning the mechanical factors influencing lubrication should also be secured. Some of these items are as follows:

1. Cylinder diameter and length.
2. Piston stroke.
3. Piston diameter and length.
4. Area of piston in contact with cylinder wall.
5. Piston clearance. If piston is tapered, the clearance at all points of variation should be ascertained.
6. Number and type of piston rings.
7. Width of piston rings.
8. Piston-ring pressure.
9. Character of piston metal.
10. Diameter and length of all bearings—connecting-rod, wristpin and mainshaft bearings.
11. Bearing clearance.
12. Type of bearing metal.
13. Method of lubrication (full description including lubricators, oil grooves, pipes, leads, pumps, filters, capacity of system, rate of circulation, etc.).
14. Method of fuel admission (carburetor, air injection, solid injection).
15. Method of fuel ignition (spark, hot plate, compression).
16. Type of cycle (two-stroke, four-stroke).
17. Method of cooling engine (air, forced-water circulation, siphon, oil, etc.).

Amount of Lubrication.—The lubrication consumption varies over wide limits in various makes of engines. Table XXXII gives values that follow very closely the amounts necessary on engines from 200- to 600-hp. capacity.

TABLE XXXII.—DIESEL LUBRICATION REQUIREMENTS

	DROPS PER MINUTE
Air compressor cylinder	3- 4
Engine cylinder	25 30
Wrist pin	10 15
Crank pin	15 25
Helical gears	5 10
Governor	4- 8
Exhaust valve stem	2 5
Admission	2 5

The engine as a whole should use about 1 gal. of oil for each 3,000 or 4,000 hp.-hr. of engine rating. This means that a 1,000 hp. engine should run 3 hours on 1 gal. of oil even though the load is only 500 hp.

Reclaiming Lubricating Oils.—The methods of reclaiming the oil from the Diesel lubricating system may be classified as follows:

Type I. Tanks in which the oil is allowed to settle.

Type II. Reclaimers in which the oil is heated above the boiling point of water in order to remove the latter; the oil after this is filtered and clarified by settling

Type III. Those systems where the oil is treated by chemicals.

Type IV. Centrifugal separators

The only advantage possessed by Type I is that of cheapness, for all that is needed is a covered tank or even a barrel. However, the time required for the water to separate from the oil and for the dirt, grit, and carbon to settle to the bottom is considerable. If the oil is disturbed by the pouring of an additional supply of used oil, the separation will be prolonged. Only in places where a more modern arrangement cannot be procured should this system be used.

Type II is the system found in many plants. This may be obtained from one of the several builders of oil purifiers or may be built locally. In Fig. 299 is shown a filter that has been used in many plants. Filtration is obtained by passing through terry cloth bags and the water is settled out by being held in the receiver for some time. This device, while satisfactory when the amount of carbon and dirt in the oil is not large, will not eliminate the water and sludge without heating. Consequently an improved design is now offered to oil-engine plants.

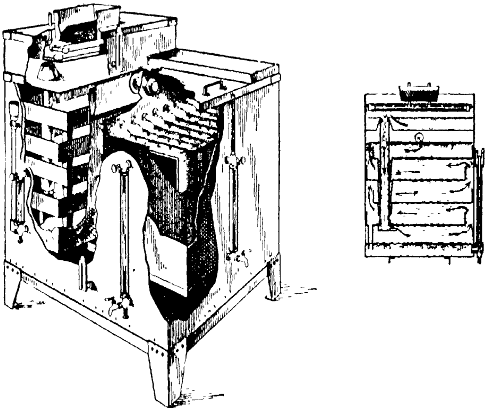


Fig. 299.—R-P filter for Diesel engines.

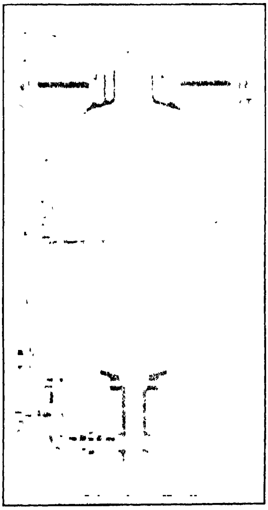


Fig. 299A.—De La Vergne oil reclaimer.

Type III covers those systems wherein the oil is placed in a tank and after being heated is dosed with a certain amount of chemicals. The chemicals cause the dirt and carbon to settle at the bottom. The oil is then drawn off to a second tank where further heating of the oil decreases its viscosity enough to allow the entrained water to settle out.

In Fig. 299*A* is shown the De La Vergne oil reclaimer using Oilite.

On analysis Oilite compound gave the following results.

Sodium Carbonate (Ha_2CO_3) 84 per cent.

Sodium Hydroxide ($HaOH$) 3 per cent together with traces of potassium, nickel chlorine and other minerals. The compound is merely a commercial soda ash and as such may be purchased at any drug shop.

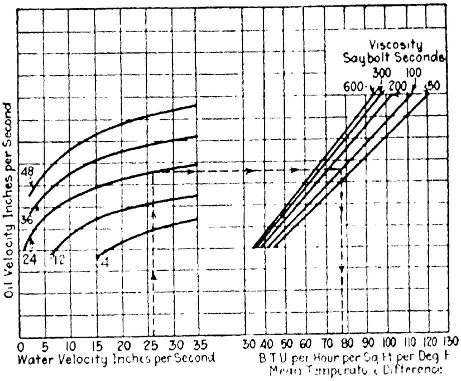


Fig. 300.—Heat transfer in oil coolers.

Type IV, the centrifugal filter, is nothing more or less than a cream separator. The dirt and carbon, as well as the water, is thrown out of the oil by centrifugal force. The results are absolutely satisfactory and the oil is restored to its original grade. If the service is particularly difficult the oil may subsequently be put through a filter press or terry cloth filter.

Oil Coolers.—Practically all Diesels have coolers of the shell and tube type for the purpose of cooling the oil after use. The size, etc., are determined by the builder. However in Fig. 300 is a chart by S. E. Derby as published in Power, showing the coefficient of heat transfer in oil coolers. The viscosity of the oil is of importance, it will be noticed. Figure 300*A* shows the temperature drop with a variation in the oil flow and with several water-oil ratios.

Lubrication of Air Compressor.—The most common cause of compressor valve leakage is carbonized lubricating oil. This forms a hard crust around the valve and on the seat, and in cases where an engine is operating in a dusty atmosphere a large percentage of gritty dirt mingles with this carbon. Most of the lubricating oils used in air compressors are corrosive under operating conditions, and it is not uncommon to find valves that have the appearance of having been etched with acid. This corrosion usually starts at one edge of the valve face and gradually reduces the area of contact between valve and seat, so this action may proceed for quite a long time before contact is broken at any one point and leakage starts.

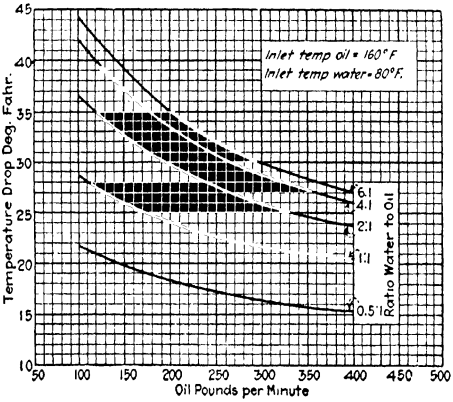


Fig. 300*A*.—Temperature changes in oil cooler.

Most air compressors get too much lubricant, either through too liberal adjustment of lubricators or by inhaling oil from the crankcase. The first can be corrected after the proper amount of oil feed is determined by trial, but the second is not entirely within the control of the operator. If the first-stage cylinder is next to the crankcase, the vacuum in the cylinder during the suction stroke draws some of the oil fog from the crankcase up into the cylinder. In three-stage machines it is customary to place the second-stage cylinder next to the crankcase so that there will always be considerable pressure opposing the entry of oil, but even with this arrangement a surprisingly large amount of oil will work up into the cylinder. As further preventives the piston next to the crankcase should be provided with an oil-scraper ring near its crankcase end and a splash hood should be built around the crank, hugging the connecting rod as closely as possible to prevent oil being thrown from the crank up into the cylinder when near top center.

A part of the oil in the compressor cylinders will be vaporized by the highly heated air and is carried into the intercoolers, where it is condensed and can be blown out through the drains. The part that does not vaporize is carried through the valves and deposited in the air pipes beyond the last stage. If leaky valves or failure of cooling water causes the air temperature to rise, it may get high enough to ignite the air and oil mixture in the pipes and cause a serious explosion. That portion of the oil that is evaporated and recondensed in the intercoolers may not all pass off through the drains; some of it may stick to the tubes in the cooler, and in time the coating of oil on the tubes will seriously reduce the cooling capacity and cause a very high temperature rise in the air. Whenever the engine is shut down for an extensive overhaul, the air compressor and all air piping and tanks should be steamed out if a steam supply is available, or blown out by air. By removing all valves the steam may be passed through the whole system, and if it is left on for at least twenty-four hours a surprising amount of oil can be drained out.

CHAPTER XX

AIRLESS-INJECTION OIL ENGINES

Solid Injection. Gas Injection

General.—A few years after the Diesel engine using high-pressure air as the means of forcing the oil into the cylinders had become a demonstrated success, engineers turned their attention to the building of an engine without the air injection. It was considered that the air-injection system was somewhat cumbersome and that the air compressor caused a large percentage of the Diesel operating troubles. That high compression was necessary both for efficiency and to obtain auto-ignition of the fuel was generally conceded. Likewise the highly satisfactory atomization and turbulence obtained by the use of an air spray compared with the results from direct pump injection of the oil made the retention of the air injection desirable.

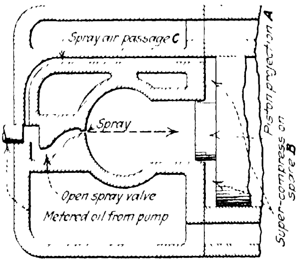


Fig. 301.—Hasselwander engine.

The Hasselwander Engine.—As a consequence one of the first departures from the true Diesel was Hasselwander's Engine. In this machine the combustion chamber of which appears in Fig. 301, the air compressor was eliminated. To provide the injection air the piston was provided with a plug as indicated. When the piston approached head dead-center on the compression stroke, the plug entered the passage between the cylinder and combustion chamber. A certain amount of air was thereby trapped in the cylinder space. Since the clearance between the piston and cylinder head was very small further movement of the piston raised the pressure of this trapped air much beyond that existing in the combustion chamber. This highly compressed air flowed through the small passage to the open-type oil-spray chamber where the air picked up the fuel and sprayed it into the combustion chamber. It will be noticed that here there was, at least in theory, an injection without the air compressor. Operating difficulties such as leakage by the plug, too early flow of the air to the spray valve, etc., prevented this engine from being an unqualified success.

Trinkler Oil Engine.- Still another design was the Koerting-Trinkler super-compression system. Instead of the plug cutting off an air charge, as in the Hasselwander, the engine was fitted with a small piston actuated by a cam. At the proper time for fuel injection the rapid travel of the cam moved this piston, compressing the air in front of it. This air then carried the oil spray into the cylinder. This device had operating faults that prevented its general use.

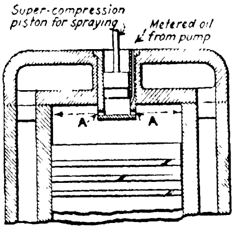


FIG. 302.—Hoflinger super-compression device.

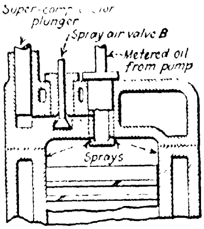


FIG. 303. Gernandt's fuel-injection design.

Hoflinger Super-compression Engine.—Still another device, that of Hoflinger, is illustrated in Fig. 302. The action of this is similar to the Trinkler and needs no explanation.

Gernandt Injection Device.—Gernandt employed a device quite like Fig. 303 but to obtain a timed injection of the oil, he placed a valve *B* between the air plunger and the fuel cup. In this way the oil was injected into the cylinder at a definite time, this eliminated the excessive preignitions of the other devices.

All of these designs made use of air injection although the compressor as a separate mechanism was absent. Consequently these cannot be termed solid injection nor even airless-injection but were modifications of the standard Diesel construction. allowing a saving in costs at some sacrifice in operating results. The compression pressures were also as high as in the true Diesel.

Classes of Solid Injection Engines.—In dividing those engines capable of using heavy oil with an efficiency almost equal to that of the Diesel it is probable that the most logical method is to classify them as gas-injection and solid-injection engines.

In the second class should be included all those engines where the oil is introduced into the cylinder or combustion chamber by direct pump action.

In the first class will be placed those engines where the fuel is introduced into some form of primary compartment where it is partly or wholly gasified and then forced into the cylinder by means of the pressure developed by a small partial combustion in this compartment. This type is by far the most numerous, although it may, in the case of some engines, be difficult to prove the correctness of this classification.

Gas Injection. Hvid or Brons Principle.—Several makes of "solid-injection" engines use the "cup" form of injection and atomization of the fuel. With this device the combustion of the atomized fuel is accomplished by means of the heat of compression, exactly as in the true Diesel engine. The sectional view in Fig. 304 shows the cup employed on the early Brons engine as built in Austria. The Hvid engine in the United States is the Brons and takes its name from the American licensee, Hvid. Incidentally the basic United States patents have expired and the only ones in force are those covering improvements, etc.

The operation of the cup shown in Fig. 304 is based on the fact that every oil, no matter how heavy it may be, contains some light hydrocarbons that will vaporize or distil at a fairly low temperature. The explosion of these lighter parts of the fuel provides the propellant whereby the remainder of the fuel is injected into the engine cylinder in a finely atomized condition. It is a further matter of common knowledge that the temperature of ignition of an oil is dependent upon the degree of atomization.

In Fig. 304 the cup or primary cylinder *A* has parts *B* and *C* that communicate with the outside air at *D* and with the fuel supply at *E*. As the piston *F* starts on its suction stroke, the pressure inside the cup *A* decreases, as the joint between the cup and the cylinder casting is not tight. The reduced pressure in the cup causes a small charge of air and oil to be drawn into the cup. The oil port *G* is controlled by the action of the governor, thereby regulating the amount of the fuel charge. At the proper point, at the end of the suction stroke, the cup moves upward and closes the fuel and air ports *B* and *C*, and at the same time covers the communicating port *H* in the side of the cup. As the piston returns on its compression stroke, the air leaking past the joint between the cup and the cylinder raises the pressure in the cup to from 350 to 100 lb. per square inch. The resulting temperature is sufficient to vaporize and ignite some of the light hydrocarbons, causing a maximum ignition pressure inside the cup of about 700 lb. per square inch. Just before the crank reaches dead-center, the cup is rotated a slight amount, uncovering the port *H*. The extremely high pressure in the cup immediately forces the heavy oil charge out through the port *H* into the cylinder space. In passing through the port into the low pressure existing in the cylinder, the oil is atomized sufficiently to unite with the air charge in the cylinder. It will be observed that the character of the oil determines in a great measure the time of ignition and the shape of the combustion line on the indicator card. If the oil is extremely heavy, with the lightest hydrocarbons of fairly low gravity, then the light portion will

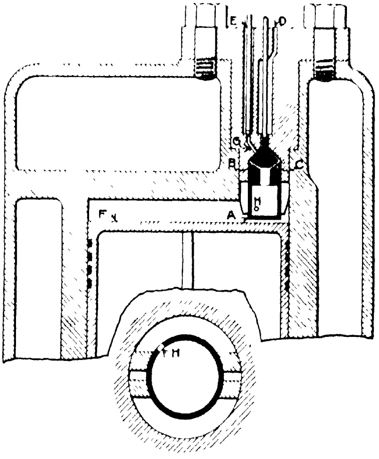


FIG. 304.—Early form of Hvid ignition device.

not ignite in the cup until practically dead-center is reached. The pressure in the cup, due to this primary combustion, will not reach a high value before the port *H* is opened. The pressure difference existing between the cup and the cylinder will not be great, and consequently the injection through the port *H* will be slower, and the combustion line on the card will be practically horizontal.

This ignition device was employed on the St. Mary's oil engine of the vertical type when first built in this country.

Modern Hvid Precombustion Cup.—It was found that it was impossible to have a gas-tight fit between the cup and the lugs.

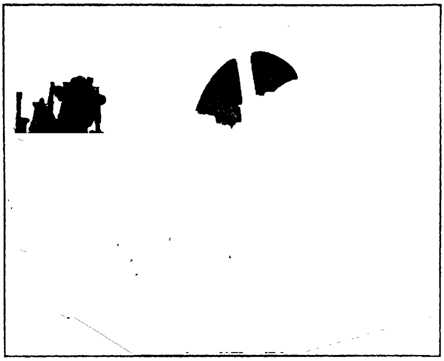


Fig. 305.—St. Mary's Hvid engine.

In consequence the oil was sprayed out through the ports as soon as the primary ignition raised the pressure above the cylinder pressure.

The rotation of the cup was early abandoned and regulation of the time of the oil injection into the cylinder was obtained by the use of holes of the proper size and the correct compression pressure.

St. Mary's Oil Engine.—The horizontal St. Mary's engine, Fig. 305, makes use of the Brons cup of the stationary type.

Lyons-Atlas Hvid Engine.—The cup used on the Lyons-Atlas Hvid engine (now no longer built) is the same type as that on the St. Mary's engine, Fig. 305. A cross-section of the Lyons-Atlas (Midwest Engine Co.) cylinder is shown in Fig. 306. Here the cup is equipped with a fuel valve which is mechanically operated from the camshaft. In operation a charge is drawn into this cup by atmospheric pressure when the engine is on the suction stroke. The fuel valve controls the timing of the period of fuel admission. To regulate the amount of fuel admitted, the needle valve *E* is connected to the governor. The movement of the governor, through a suitable linkage not shown in Fig. 306 rotates the needle-valve stem; this, in turn, alters the area of the fuel port or passage *C* and so controls the amount of oil flowing from the overhead tank during the time the fuel valve is open. It is to be observed that the fuel charge enters the cup on the suction stroke of the engine. It remains here during the compression stroke until the temperature is sufficient to ignite the fuel.

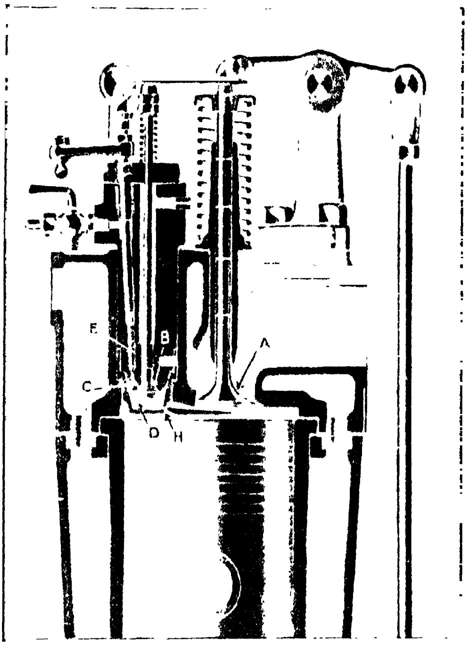


FIG. 306.—Lyons-Atlas Hvid engine.

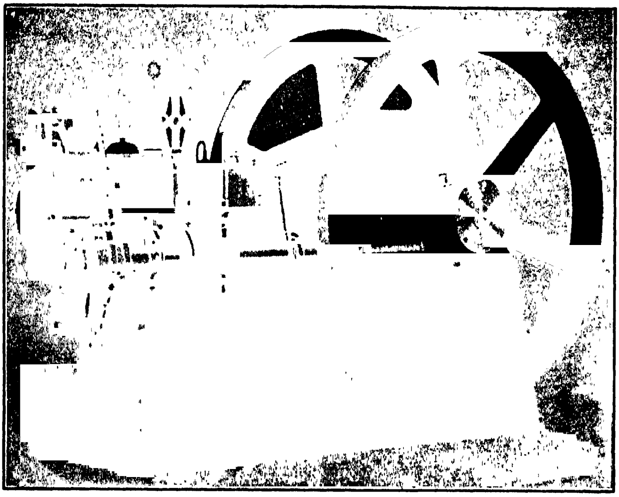


Fig. 307.—Midwest V.D.H. engine.

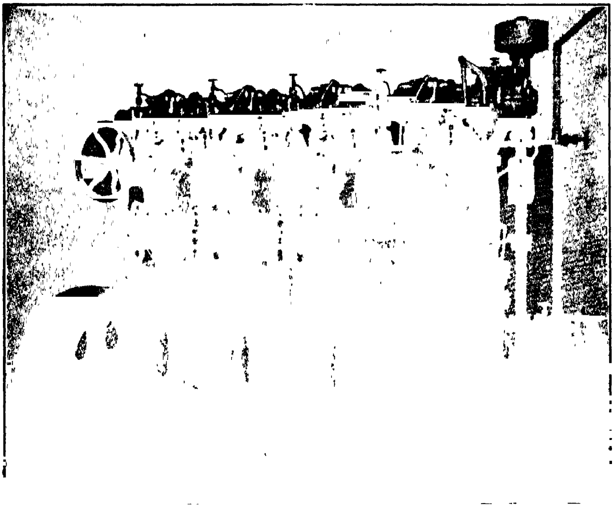


Fig. 308.—Midwest vertical engine.

An illustration of the V.D.H. engine appears in Fig. 307. The governor, which is of the spring-loaded type, is driven off the layshaft and controls the fuel-needle valve *E* in Fig. 306.

Lyons-Atlas Vertical V.D.H. Engines.—On their vertical V.D.H. engine the Lyons-Atlas Co. eliminated the governor-controlled needle valve and used a fuel pump for each engine cylinder, the fuel pump being under governor control.

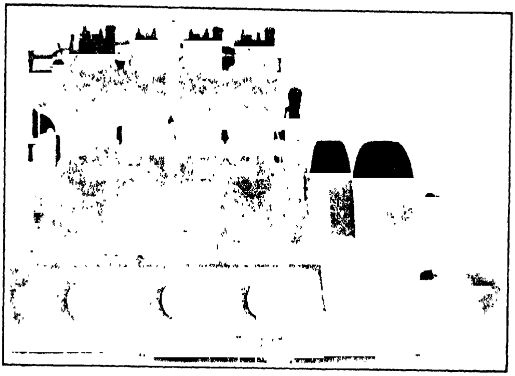


FIG. 309.—Dodge Hvid engine.

The vertical engine is illustrated in Fig. 308. The cylinders, which are 9-in. bore by 13-in. stroke, are mounted on A-frames. On one side the frames are open, being supported by tension rods. The openings are covered by steel oil guards that are removable. The cylinder construction is quite unusual. Each cylinder is provided with a removable skirt. This skirt can be unbolted, allowing the piston to be swung out to the side without unshipping the connecting rod from the crank. The engine is no longer built.

Dodge Hvid Engine.—The Dodge Manufacturing & Sales Co., build a vertical oil engine with the Hvid ignition device. The engine is shown in Fig. 309. This engine has each cylinder and its crankcase cast in one piece, the several units being bolted together to form the engine.

The engine is built with one size of cylinder, a 12½ hp., and the capacity ranges from 12½ to 75 hp., Table XXXIII. The crankshaft is of open-hearth steel forged from a high-carbon steel billet, and heat-treated, diameter 3½ in., length of end main-bearing 6 in., center bearing length 5 in., crankpin 3½ in. diameter by 3¾ in. in length. The main bearings are babbitted cast iron shells fitted into the machined bedplate and can be rolled out without removing crankshaft. The crankpin bearings are heavy bronze boxes babbitt lined.

Table XXXIII.—Dodge Oil Engines

Size	Number, cylinders	Net weight, lb.	Domestic shipping weight, lb.	Boxed for export	
				Gross weight, lb.	Measurements, cu. ft.
12½ b.hp.	1	3,000	3,400	100	6
25 b.hp.	2	4,000	4,400	3,900	90
37½ b.hp.	3	5,100	5,600	5,000	110
50 b.hp.	4	6,200	6,700	6,200	130
75 b.hp	6	8,400	9,100	7,500	150
Exhaust silencer	..	500	600	10,000	190
Fuel service filter tank	..	35	35	700	20

The connecting rod is an open-hearth steel forging machined all over. The piston-pin end is slotted out of the solid to receive an adjustable box of high grade bronze. The two halves of this box are adjusted and held firmly clamped together by means of a steel wedge in the upper end of the rod. The crankpin box is of the marine type and consists of heavy bronze halves, babbitt lined.

The trunk piston is fitted with five concentric rings and one oil ring at the bottom. The piston pins are of hollow steel tubing, hardened and accurately ground to size. One end has a diameter slightly smaller than the pin proper to allow for regrinding after wear. Each end is provided with a key to prevent turning in the piston bosses.

The camshaft is located inside of the crankcase. By taking off the frame side cover plates, it becomes accessible for its whole length, and can be removed as a unit in a few minutes.

The cylinder head, Fig. 309*A*, is uniform and symmetrical owing to the arrangement of the valves and there is ample room for cooling water between the valves and especially around the exhaust valves.

The governor is of the Pickering type and is driven from the crankshaft through a helical gear and two short shafts connected by bevel gears. The oil is measured by the governor adjusting the cup regulating valve according to the load and speed of engine.

The cooling-water pump is bolted to the end cover on the flywheel end of the engine and is readily accessible. It is driven by an eccentric off the intermediate shaft running at half engine speed. All parts touching water are of bronze. Circulating water is lead to the bottom of each cylinder jacket and passes from the top of these through elbows to the cylinderhead. The capacity of the pump allows it to carry excess water so as to secure ample cooling of the cylinders in hot climates.

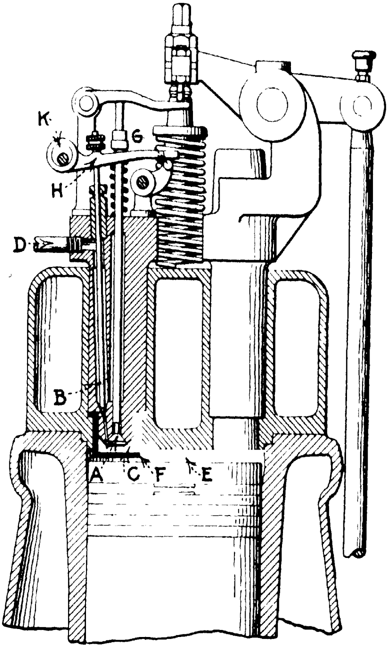


Fig. 309A.—Cross-section of Dodge engine.

An air compressor is for starting air and is mounted at one end of the frame and is entirely enclosed. It is capable of compressing up to 250 lb. pressure and filling the air-starting tank in a few minutes, after which it can be unloaded.

A mechanically operated force-feed multiple oiler supplies all cylinders, main bearings, connecting rods and camshaft bearings. Crankpins are lubricated from the force-feed lubricator by means of oil rings. The amount fed to each bearing may be observed and accurately adjusted, for quantity of oil delivered.

Hercules Hvid Engine.—The Hvid engine is built in small powers—1½ to 10 hp.—by the Hercules Gas Engine Co. This engine is likewise sold by Sears-Roebuck Co. under the trade name of the Thermoil Engine, Fig. 310.

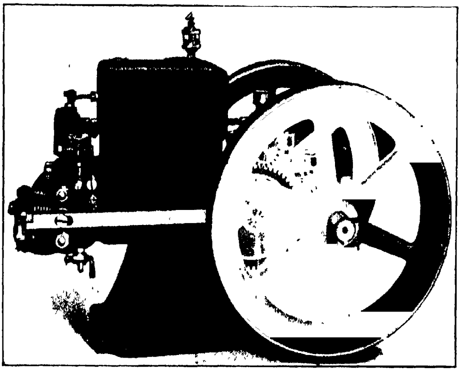


FIG. 310.—Hercules engine.

Jones Hvid Engine.—Similarly the small-powered Hvid engine is sold by the Jones Oil Engine Co.

Petroleum Engine.—A considerable number of small Hvid engines have been exported by the Petroleum Engine Co. to the South and Central Americas.

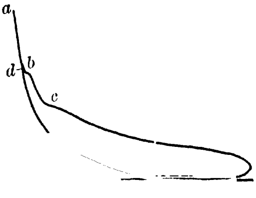


FIG. 311.—Indicator diagram of Hvid cup pressure.

Operation of Hvid Engines.—It has been claimed by certain engineers that there is never a primary ignition in the cup. These men claim that the spray effect is obtained solely by the pressure difference between the cup and cylinder after the piston has reversed its stroke. The motion of the piston would cause the cylinder pressure to drop sharply. However, indicator diagrams taken from the cup show pressures as high as 700 lb., indicating the existence of the cup ignition, see Fig. 311.

Care of Hvid Cup.—Since the injection is not positively controlled, the openings in the cup must be altered in size to regulate the flow of oil through them. On a light oil, if the openings are large, the injection will be early, causing preignition. The remedy is to peen the holes. The compression carried, averaging 450 lb., requires a sturdiness of construction approaching the Diesel. The cup and fuel valve must be maintained in the best of shape. The smallest leak will lower the cup compression. Since the explosive pressure approximates 600 lb., it is evident that the valve must be ground often and with the utmost intelligence. As there is no water-cooling of this valve and seat, it will corrode and pit if it does not receive attention.

It has been the general experience that there is no great difficulty in building and in operating a single-cylinder Hvid engine. In larger powers, where more than one cylinder is required, many have found that the engine behaves at times in a most unsatisfactory manner, either running at an irregular speed or at a greatly reduced capacity.

The engine trouble may be traced to one or two faults. In the first place the size of the passages through the cup largely determines the length of the combustion period. If they are not the same in each cup, the rate of injection will not be the same for the two cylinders. The combustion will not be uniform and the speed may vary. In a single-cylinder engine, if the passage is too large, the engine behavior will allow the trouble to be detected and remedied by peening the holes to a smaller diameter, or enlarged if the combustion proves to be so delayed as to cause the exhaust temperature to be excessively high. In case of multi-cylinder units the trouble is not so easily detected.

The chief operating trouble is caused by carbonization of the cup. In the single-cylinder engine this is detected at once by the loss of power. In case of a two- or three-cylinder unit the detection is difficult by reason of a certain characteristic possessed by this type of engine. The engine will develop an extraordinary power from a given cylinder size. If the load is heavy, the engine will hang on much like a steam engine, giving an increased torque even through the speed drops. If the cup of one cylinder carbonizes, the other cylinders will carry the load at reduced efficiency and with a rapid carbonizing of the cups of these cylinders. One cylinder may be put entirely out of commission before the engine shows by its behavior that anything is wrong.

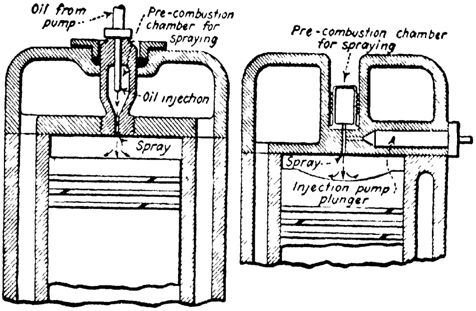
By drilling inspection holes in the exhaust elbow at each cylinder, the operator may detect at once, both by the sound and by the smoke of the exhaust, if one cylinder begins to lag. As soon as this occurs, the engineer should cut off the fuel supply to the cylinder in question, easing up on the load if necessary. The repeated compressions and re-expansions of the air drawn into the cylinder cause carbon in the cup to combine with the oxygen and burn. Usually, twenty to twenty-five compressions will completely clear the cup and the cylinder may then be put back into service. With this arrangement the multi-cylinder Hvid engine will function as positively and reliably as any other type of oil engine.

Fuel Consumption. Hvid Engines.—The fuel consumptions of the various engines employing the Hvid principle are practically the same. Table XXXIV covers tests on a three-cylinder vertical engine, 9½-in. bore by 10½-in. stroke, and is quite representative of this type.

TABLE XXXIV.—TEST ON OIL ENGINE EMPLOYING HVID PRINCIPLE

Test No.	I	II	III	IV
Bore of cylinder, in.	9½	9½	9½	9½
Stroke, in.	10½	10½	10½	10½
Fuel	Kerosene	Mexican fuel oil	Standard Oil Co. fuel oil	Lansing fuel
Weight per gallon, lbs.	6 9375	7 1875	7.375	6 9375
Weight per pint, lbs.	0.8672	0 8984	0.922	0.8672
Gravity, deg Baumé	42	33	29	39
Flash open flame, deg F.	160	200	196	210
R p.m.	315	330	345	340
Pounds pull net.	191	191	191	191
Brake-arm circle circum., ft.	33	33	33	33
Torque, lb.-ft.	1003 3	1003 3	1003 3	1003 5
Horsepower	60 165	63 03	65.895	64 94
Fuel, lbs. per b.h p.-hr.	0 586	0.500	0 481	0.466
Operation of engine	Smooth	Very smooth	Smooth	Smooth
Exhaust	Clear	Trace of smoke	Almost clear	Trace
Engine troubles	None	None	None	None
Maximum b.h.p. with slight smoke	60.16	85.00	84.15	86.70

Steinbecker Precombustion Chamber.—While no attempt has been made to take up European designs as a whole, still certain later-day American engines are the outgrowth of European designs. For this reason, those European inventions that are basic or the parent of modern engines will be outlined. A number of years ago a German engineer Steinbecker patented the fuel injection design shown in Fig. 312. This, the reader will observe is to all intents the Brons cup device with the mechanically-operated fuel valve eliminated. The design embraced the idea of compressing the cylinder charge of air to a pressure of 450 to 500 lb. Part of this highly heated air was forced through a narrow orifice into a precombustion cup or chamber in the cylinder head. In passing through this orifice, which was highly heated by the previous explosion, the air was heated, or in all events was not cooled. Just before top dead center the fuel was injected through a spray nozzle by the engine pressure feed pump. The nozzle action served to break up the fuel into minute drops. In contact with the air in the chamber the oil spray was vaporized. Since the air supply was limited, only a small portion of the fuel actually burned. But this primary explosion expelled the vaporized fuel out of the orifice into the cylinder. Here the oil vapor mixed with the air and combustion proceeded. Combustion was in the form of air explosions and by no means gave constant pressure combustion as in the case of the Diesel. In fact, the action was not as good as with the Hvid engine.


Oil from pump
Pre-combustion chamber for spraying
Oil injection
Spray
Pre-combustion chamber for spraying
A
Spray
Injection pump plunger

STEINBECKER PRE-COMBUSTION GAS INJECTION

LATER STEINBECKER COMBUSTION DEVICE

FIG. 312. Steinbecker injection action.

From this design may be traced a number of modern precombustion engines, including Steinbecker's later design.

Western Oil Engine.—One of the first of the precombustion engines developed in the United States was the Western, called by its builders, "The Western Diesel." This is illustrated in Fig. 313. The engine operates on the four-stroke-cycle principle and is exceedingly rugged in design. The precombustion chamber is located in the cylinder head almost in line with the cylinder bore. The piston comes up close to the cylinder head and to direct the oil vapor into the narrow clearance space between piston and head, the chamber orifice is curved as shown.

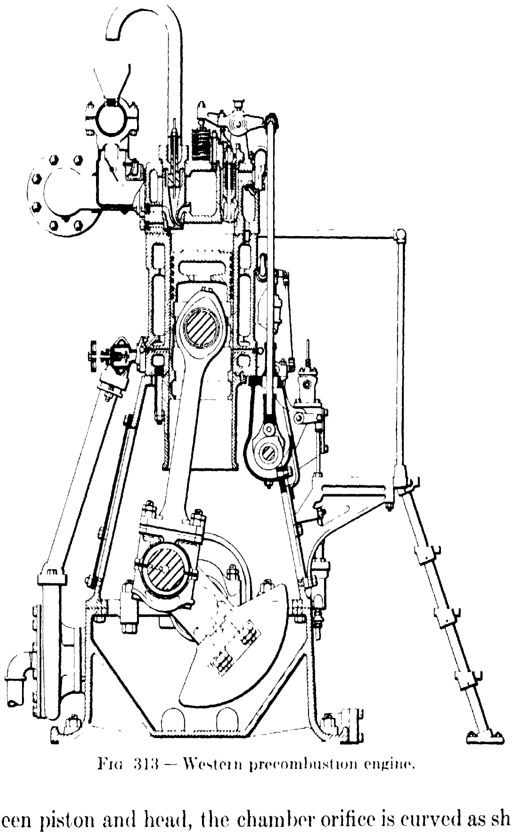


FIG 313 — Western precombustion engine.

The fuel pump is outlined in Fig. 314 at *A*. It consists of a single plunger driven by a rocker *B* and cam *C*, which is mounted on the engine-valve camshaft. The governor weights *D* by shifting the rod *E* limit the suction stroke of the pump plunger and so regulate the amount of fuel injected into the precombustion chamber.

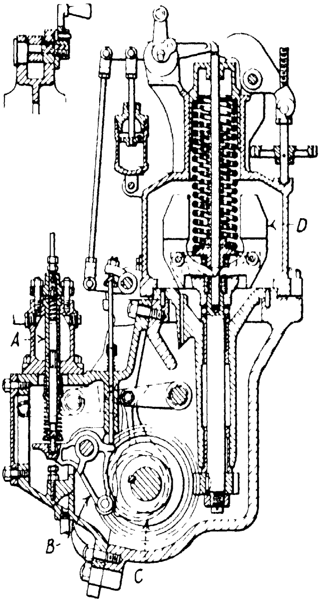


FIG. 314.—Fuel pump of Western engine.

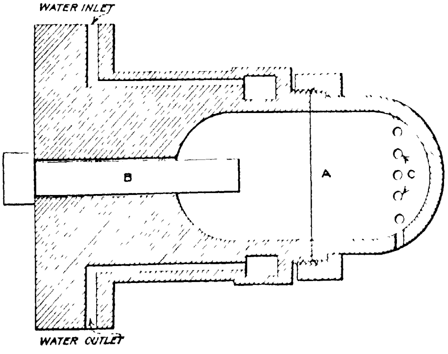


FIG. 315.—Nordberg precombustion cup.

Nordberg Ignition Device.—Another form of ignition cup, adopted by the Nordberg Manufacturing Co. for their two-stroke-cycle airless injection engine (or high-compression, as the manufacturers called it) engine, appears in Fig. 315. This, it will be observed, is a development of the idea shown in Fig. 304. The fuel oil is injected into the cup *A* through the atomizing nozzle *B*, by the action of the fuel pump, a few degrees before mid-point in the compression stroke. This fuel, as it leaves the injection nozzle, mixes with the air which has been forced into the cup by the advancing engine piston. As with the devices already discussed, part of the oil ignites as soon as it enters the hot cup and mingles with the air. The combustion of this part of the oil charge creates, within the cup, an extremely high pressure, which is much greater than the engine-compression pressure. This difference in the two pressures causes the remainder of the fuel charge, now in vapor form, to be forced out through the ports marked *C*.

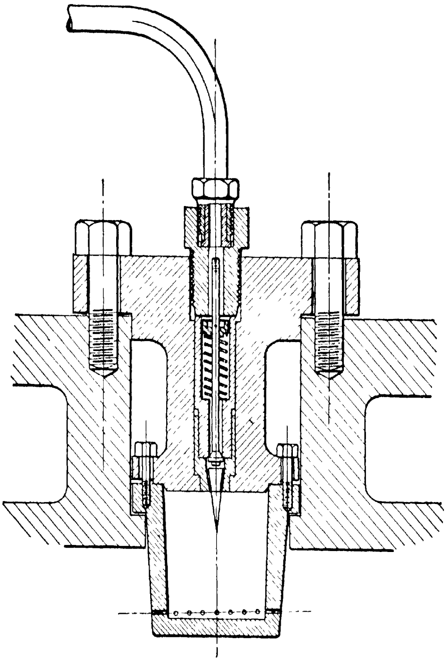


Fig. 316.—Nordberg precombustion cup, late design.

In this engine the point of injection of the fuel into the cup is about mid-stroke of the piston. Since the time necessary to vaporize the fuel is subject to variation, with certain oils preignition may occur. If the oil is of light gravity, the high-compression pressure carried will cause it to ignite very early. This primary explosion forces the main charge into the cylinder before dead center, causing preignition and bearing pounding. To avoid premature combustion, the cup is designed with a water-cooled space; this assists in keeping down the temperature of the cup and delays combustion.

Figure 316 is a later design of cup. In this the water-cooling is discarded. The injection nozzle is provided with a spring-loaded check valve and is held in position by a screwed lock bushing instead of a locking bar as was used on the former design. The bowl of the cup is bolted to the body of the igniter; this eliminates the difficulty experienced with the screwed ring collar of the previous cup.

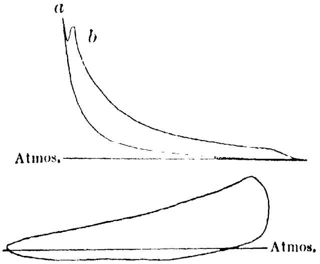


FIG. 317.—Power and scavenging diagrams, Nordberg oil engine.

Figure 317 shows an indicator card from this engine. The card reveals combustion conditions that are not along the true Hvid plan. It would appear that the primary charge did not explode until dead-center and that the piston retreated some distance before the pressure difference caused the injection of the main charge and its consequent combustion. Under such injection conditions a dull thump would be heard, produced by the burning gases striking the receding piston. If the injection had been early, as is usual, a sharp sound would have been emitted as the primary charge impinged on the advancing piston.

Fuel Pump.—The Nordberg semi-Diesel engine is equipped with a fuel pump shown in Fig. 318. The pump plunger is driven

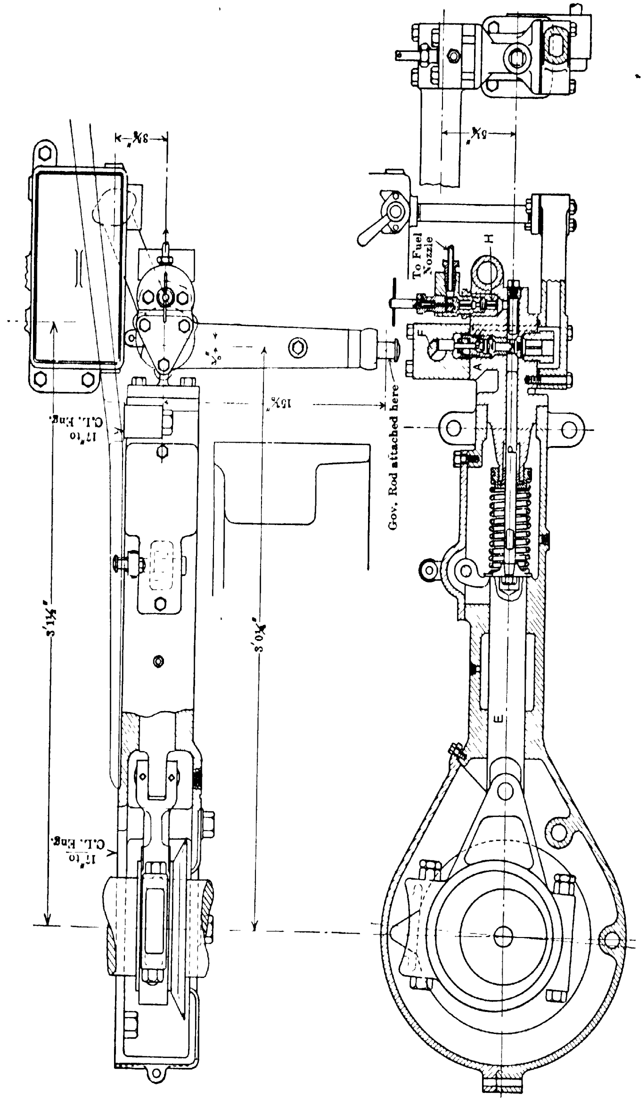


FIG. 318.—Nordberg fuel pump.

by an eccentric keyed to the engine shaft, while the fuel charge is regulated by the closure of the by-pass valve V which is under governor control. The fixed eccentric moves the pump plunger *P* on the outward stroke, drawing in a charge of oil. As the plunger reverses and moves inward on the discharge stroke, the oil, displaced by the plunger, escapes through the by-pass valve, which is held open by the cam *F*. This cam is rocked

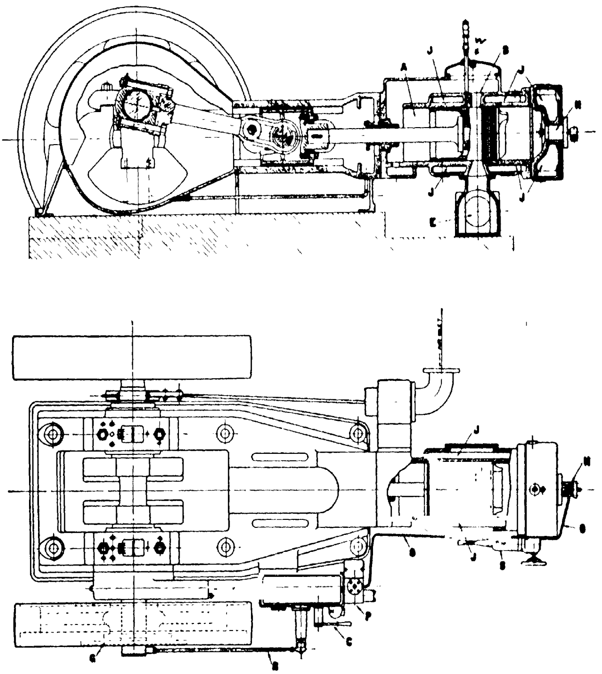


FIG. 319.—Nordberg precombustion oil engine.

through an angle by the governor eccentric rod *G*. At the proper time the cam is moved to its mid-position. The by-pass valve closes, and the oil displaced by the further movement of the pump plunger passes through the discharge valve *H* and enters the fuel atomizer.

Nordberg Engine Frame.—Figure 319 illustrates the engine section and plan. The frame is of heavy construction and has two main bearings. The piston is provided with a piston rod and crosshead. The front end of the cylinder A is closed and acts as the air compressor to furnish the scavenging air charge to the power cylinder. The air enters the compressor through the piston valve V.

Leissner Precombustion Cylinder.—Another European engineer, Leissner, brought out a design of precombustion chamber having striking features. The chamber is shown in Fig. 320.

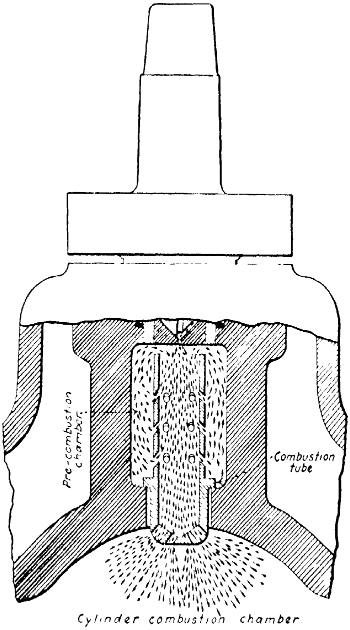


FIG. 320.—Leissner divided combustion chamber.

It was discovered by Steinbecker and other engineers that if the oil was injected into the chamber there was difficulty in controlling the primary explosion. It might cause excessive pressures if the chamber happened to be cleared of burnt gases and held only pure air. As a means of regulating the combustion, Leissner inserted a sleeve containing a number of holes into the chamber. The idea was that the oil on being injected from above would meet with the air coming in from the cylinder below. As a result of these opposing streams the oil vapor would flow through the holes into the cavity outside the tube which was filled with inert gases. Combustion could not take place until the oil vapor swept up along the tube and down the center where it mixed with the air.

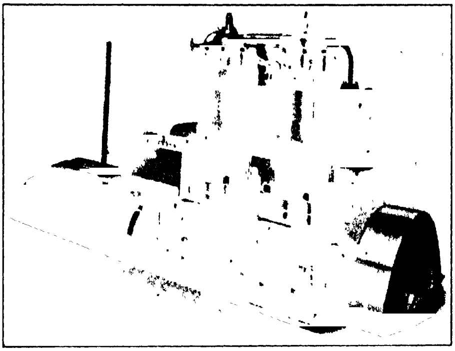


FIG. 321.—Mianus-Leissner engine.

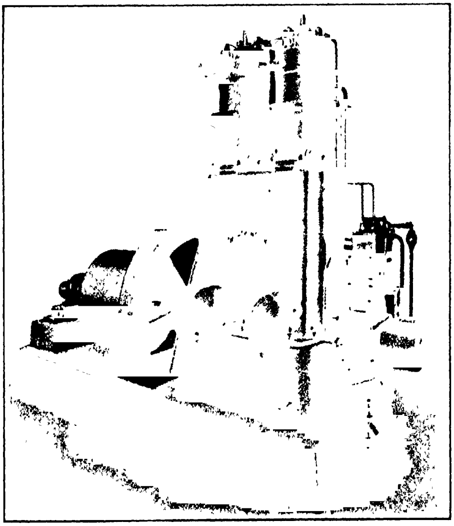


FIG. 322.—Worthington engine.

It would seem that combustion took place in the tube upon injection of the oil, for the tube burns out around the small openings, and the engine functions as well without as with the tube.

Mianus Oil Engine.—The Leissner engine is built in the United States by the Mianus Motor Co., both in Marine and stationary units. The horsepowers of these units are low, ranging from 9 to 30. The fuel combustion is approximately 0.45 lb. per b. hp.

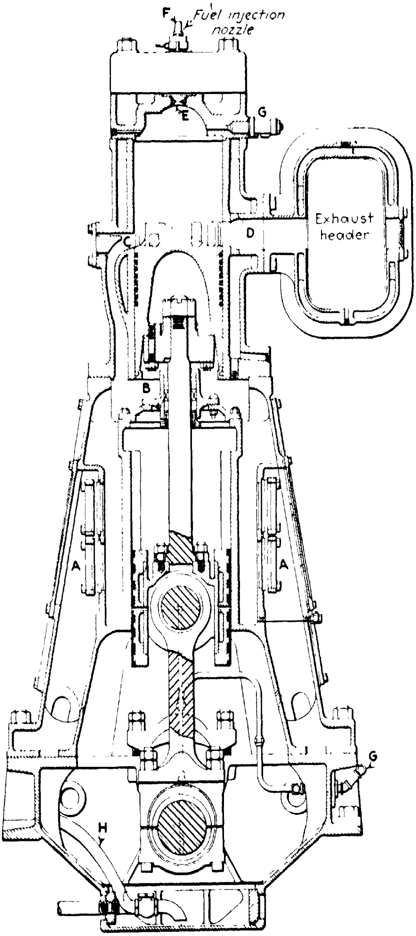


FIG. 323.—Cross-section Worthington gas-injection engine.

Worthington Precombustion Engine.—The Worthington Pump and Machinery Corp. has built Diesel engines since 1912. For small powers a precombustion engine, or as the builders call

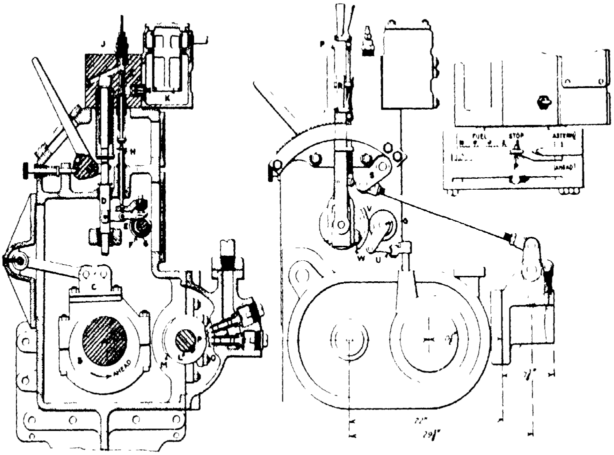


FIG. 324. Fuel pump of Worthington engine.

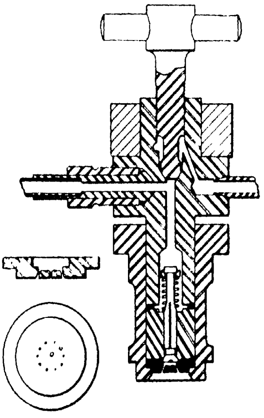


FIG. 325.—Worthington fuel-spray nozzle.

it a "solid-injection-Diesel," was developed at the Cambridge works. As originally designed, the engine was provided with a tube and ring in the precombustion cup, with the idea of limiting the amount of primary explosion. It was found, however, that the tube served no useful purpose save possibly to insure ignition with low pressures. The tube was uncooled and after the engine was started by using a "punk" stick, the hot tube made the primary explosion a surety. The uncooled tube proved to be subject to destruction. Consequently, the tube idea was abandoned and as the engine is now built the precombustion device is nothing more or less than a cavity in the cylinder head.

The Worthington engine is shown in Figs. 322 and 323. The fuel pump is illustrated in Fig. 324. It consists, essentially of a simple plunger pump provided with suction and discharge valves. The engine shaft is fitted with eccentrics, and a roller on the eccentric strap strikes its respective pump plunger. The plunger has a lever fulcrumed on a cam controlled by the governor. As the plunger moves upward, the lever strikes the shank of the suction valve, thereby lifting the suction valve and so permitting the oil to spill back. The amount of oil forced out of the discharge valve depends upon the time at which the suction valve is lifted. This is controlled by the governor which shifts the cam already noted. The spray nozzle is shown in Fig. 325.

Solid Injection Oil Engines

High-pressure Solid Injection Engine.—Vickers of England, before the late war, brought out a high-pressure oil engine in which the oil was injected by pressure instead of by air. The fuel or spray valve was provided with a needle as are all Diesels, and the needle lift was such that the fuel was introduced at a slow rate, giving almost pressure combustion. To meet the extremely high pressure requirements, the oil was pumped into an accumulator arrangement, and this maintained a constant pressure on the oil going to the fuel valve. The accumulator was made on the principle of a bourdon tube, the expansion of the flattened sides holding the pressure uniform.

This had the advantage of simplifying the engine, but it is difficult to regulate the needle valve timing. If the fuel injection is too early the engine pounds; if late, the exhaust is very smoky. Since the engineer tries to avoid early timing, it is by no means unusual to find these engines very smoky by reason of the timing being set late.

Taylor Oil Engine.—The plan of having mechanical injection with high compression pressures has been adopted by the Taylor

Machine Co. in the United States. The Taylor engine is shown in Fig. 326. The machine is two-cycle with the pistons stepped to make the scavenging pumps. It is necessary to have the engine built in units of two cylinders or multiples thereof. The air is drawn into the scavenging pump *A* from the suction manifold, as indicated by the arrows. As the piston moves upward, this air is compressed and passes out through *B*, through the transfer port *C* into the adjoining working cylinder. The burnt gases are exhausted through *D*.

The oil is pumped by a plunger pump into the fuel oil which is kept at a constant pressure by means of a relief valve. The fuel valve opens inwardly when the rocker arm strikes the stem. The duration of opening is controlled by the governor which shifts the eccentrically-turned rocker fulcrum. The fuel valve is unique in that it is provided with a ratchet device whereby each time the valve opens it is revolved about $\frac{1}{16}$ of a turn. This makes the valve self-grinding so the builders claim.

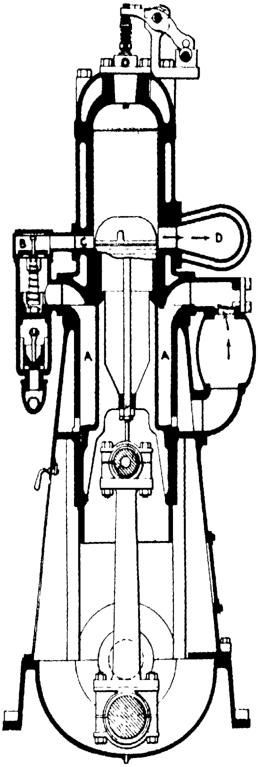


Fig. 326.—Taylor solid-injection engine.

Solid Injection Medium Compression Engines

Self Ignition.—Engineers have become accustomed to high-compression pressures in the Diesel engine. The general statement that 450 to 500 lb. per square inch compression pressure is necessary to insure self-ignition of the fuel is accepted without question. It is surprising to some to know that solid-injection oil engines, using no hot surfaces or other means of fuel ignition, are able to create a temperature above the igniting point of fuel oil with a pressure of but 300 lb.

In both the air-injection Diesel and the solid-injection oil engine the ignition of oil is brought about by the high temperature of the air charge in the working cylinder at the end of the compression stroke. The final temperature with different compression pressures and with various initial temperatures is shown in Fig. 327. In all the curves the initial or suction pressure is assumed to be 15 lb. absolute, or 0 gage, while the initial temperature T is as shown. If a gas is compressed in a non-conducting cylinder, the process is said to be adiabatic, and if the gas is pure air the final temperatures are found by following the full-line curves. The exponent n for adiabatic compression of air is approximately 1.41, but in the Diesel engine there is a considerable loss of heat to the cylinder walls and the final temperature is lower than it would be if the value of $n = 1.41$ applied. From a study of compression and expansion curves it seems that n ranged from 1.2 to 1.5, averaging about 1.35. Individual cylinders, however, show different values of the exponent. As example, one Diesel engine gave the following average values:

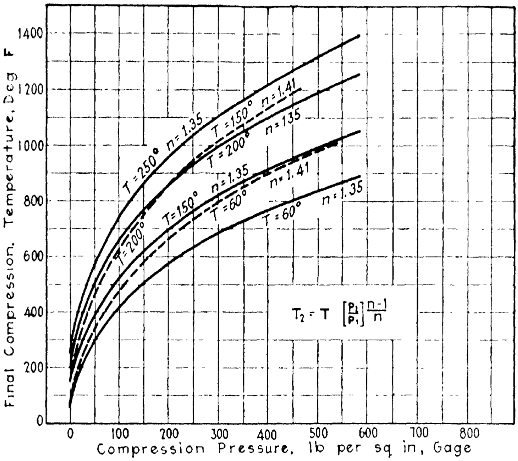


Fig. 327.—Relation of temperature to compression pressure.

Value of n

No of Cylinder	Compression	Expansion
1	1.396	1 67
2	1.371	1 207
3	1 344	1 183
4	1.378	1 244

The broken-line curves in Fig. 327 give the temperatures when the exponent *n* is taken as 1.35.

In the air-injection Diesel the pressure ranges from 480 to 520-lb. gage, while in the solid-injection oil engine compression pressures range from 280 to 450 lb. gage, depending upon the engine design. Upon examining Fig. 327, it will be found that, assuming an initial or suction temperature of 60 deg. F., a Diesel compression pressure of 480 lb. gage would cause the air at the end of the compression stroke to attain a temperature of 815 deg. If the solid-injection engine has the same heat loss, so that the exponent *n* is 1.35, a pressure of 300 lb. would give a temperature of less than 700 deg. F.

Compression Pressure Required.—The question is how, in the solid-injection engine, the temperature of the air charge can be high enough at 300 lb. to ignite the fuel, while the Diesel requires 480 lb. compression to insure ignition. The explanation usually offered is that the injection air which blows the oil charge into the Diesel cylinder causes a refrigerating effect by expansion from its original pressure of 900 lb. to the cylinder pressure of 480 lb. In the solid-injection engine no injection air is used since the fuel is injected by the direct action of a pump, and, so it is claimed, no refrigerating effect occurs. A lower compression pressure will then give a temperature sufficiently high to ignite the oil.

But the temperatures at 300 and 480 lb. air, with 60 deg. suction air, are 680 and 815 deg. respectively, a difference of 150 deg., if the same exponent of 1.35 be assumed. The question naturally arises, will the cooling effect due to the free expansion through the spray valve cause the entire cylinder charge of air to suffer a temperature drop of 290 deg.? With the injection air at 200 deg. F. and 900 lb. pressure the temperature drop in the injection air, assuming no loss of heat, or free expansion, will not exceed 50 deg. F. On the basis that the injection-air weight is 8 per cent of the weight of the air charge, the lowering of the air-charge temperature would be less than 5 deg. If it is assumed that the heat equivalent of the work of forcing the oil, which is at a high pressure, into the cylinder, where a lower pressure exists, is lost, then the adiabatic temperature drop of the injection air will not exceed 120 deg., reducing the cylinder-air temperature by some 10 deg. Obviously, if the temperature drop occurs throughout the cylinder-air volume, the cooling will be insignificant. Even if injection air and oil spray alone experienced the drop in temperature, its effect on ignition would be insignificant, for each particle of oil that by its cooling effect is at, say, 100 deg. F., is at once brought into intimate contact with a mass of air at 830 deg. F. The air-charge temperature would not drop lower than 780 deg. even if the entire mass of oil were injected instantaneously. In the actual operation ignition in a Diesel takes place before the entire oil charge enters the cylinder.

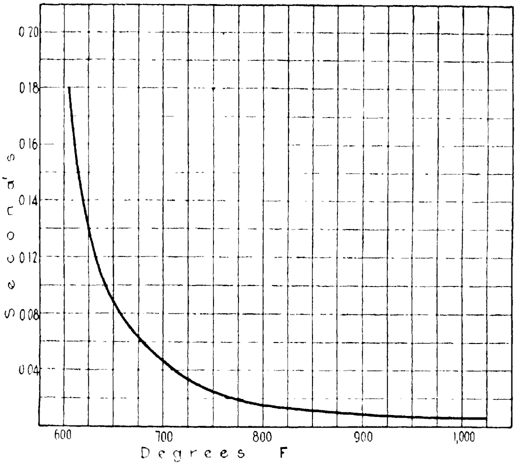


FIG. 328.—Time lag in oil ignition.

This temperature drop is negligible and does not compare with the decreased temperature resulting from leaky and worn pistons. It is extremely doubtful if this so-called air refrigeration has any material effect on the temperature required in the engine.

The higher temperature required by the air-injection Diesel as compared to the solid-injection engine, may be traced to the method and time of fuel injection, to the vaporizing of the oil after introduction and to the difference in the heat loss during compression.

When a substance such as water or oil changes its state from that of a liquid to that of a vapor, a certain amount of heat is required to raise the temperature of the liquid to the temperature at which it will boil at the existing pressure. The specific heat of oil is about 0.43 B.t.u. per pound per degree of temperature rise. After the boiling temperature is reached, before the liquid will vaporize, a certain amount of heat, depending upon the pressure and upon the character of the substance, must be added. This, the latent heat of vaporization, is in the case of oil about 170 B.t.u. When the oil reaches the boiling temperature, no further rise of temperature occurs until this latent heat has been added, whereupon the vapor continues to absorb heat at a rising temperature until combustion occurs. The higher the temperature of the cylinder air the faster will be the ratio of boiling or vaporization and the shorter will be the time interval between fuel injection and ignition. In Fig. 328 is a curve showing the time lag found by Commander Hawkes when using shale oil. There is every reason to assume that this fairly approximates the results to be expected from fuel oil.

Requirements to Insure Ignition.—It appears that the oil engine's compression pressure may be lower than that usual in a Diesel, but if so, the fuel must be introduced sufficiently long before dead-center to allow evaporation and ignition at or closely after dead-center, and the heat loss to the iron walls must be reduced. There is a pratical limit to the permissible time given the oil to vaporize before dead-center. If the engine operates at 180 r.p.m. or covers 1,080 deg. of the crank circle per second, a time interval of 0.02 sec. means 21.6 deg. This is as early as it is advisable to inject the oil. This being true, the cylinder temperature to produce ignition with this time interval must be 780 deg., as shown in Fig. 328. Turning to Fig. 327 it will be noted that with 300 lb. compression and with a heat loss corresponding to the Diesel loss an engine will give a final temperature of only 680 deg. This is less than required, and it is evident that the rate of heat loss during compression must be reduced in order to obtain 780 deg. final temperature. With an exponent of 1.41, the same as for air, the temperature at 300 lb. will be close to 780 deg. It is obvious, then, that if the solid-injection is to function, not only must the oil be introduced early to permit vaporization, but the compression must be nearer adiabatic than is the compression of a Diesel, and in addition the oil spray must not come into contact with the cool cylinder walls which will condense the spray, causing a missed ignition.

The compression curve on an indicator diagram from a solid injection engine shows that *n* is about the same as in case of the

Diesel, ranging from 1.25 to 1.37. This would seem to indicate that the above contention falls to the ground. However, the mass of air in the engine cylinder is not uniform in temperature. Part of it is much higher than that portion along the walls. In the Diesel the flat shape of the combustion chamber gives maximum cooling surface with minimum volume and the temperature will be fairly uniform. It is undoubtedly true that the center core of air in the solid-injection engine is of a temperature much greater than that of the Diesel.

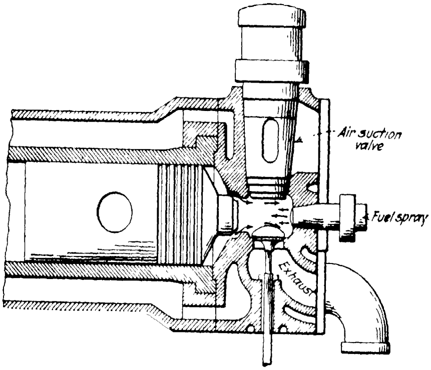


FIG. 329.—Deutz solid-injection engine (Otto Engine Co.)

Deutz Solid Injection Engine.—In the Deutz Solid Injection Engine, built in the United States by the Otto Engine Co., to limit the cooling loss, the combustion chamber is made as compact as possible to give the greatest volume with the least surface area. In Fig. 329 the center of the chamber holds a mass of highly heated air into which mass the oil spray is injected. The oil spray is by no means so thoroughly broken up as in case of air atomizing. To obtain mixing of the air and oil, the piston is provided with a boss which fits fairly closely into the combustion chamber neck. The result is that a conical ring or stream of hot air is forced up into the combustion chamber just before dead center. This completes the mixing and adds the extra heat needed to ignite the oil.

It will be noticed that the piston of the Deutz engine has a plug on the piston. Just what the plug does is the subject of controversy. It undoubtedly does cause turbulence or agitation of the air charge and this is essential to any engine. How essential is turbulence was demonstrated by Clerk in his investigation of the explosive pressure in gas engines. He found that if a gas engine was allowed to compress a charge of air and gas a few times without the ignitor being connected up, the pressure when ignition was permitted was lower than usual. It was claimed that this startling result was due to the mixture losing its turbulence or whirling motion. As a consequence, when ignition took place, the burning was slower and less complete.

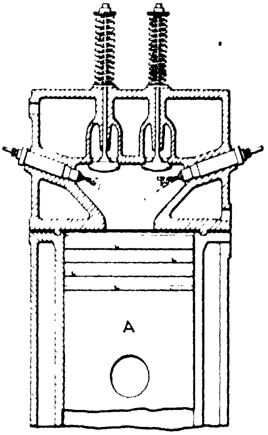


Fig. 330 Price combustion chamber

The plug is by no means necessary to obtain turbulence for the whirling of the air charge drawn in during suction continues until combustion takes place. This is proved by the results with engines not using a plug.

The plug evidently causes a stream of hot air to pass up along the cold walls. If any of the oil spray travels beyond the hot core, the stream of hot air will pick up this oil. Contact of the oil with the cold walls is thereby prevented; experiments have demonstrated that the minute an oil spray strikes a cold wall it will condense and this oil will not burn but will form carbon. The exhaust will then be smoky.

As stated heretofore the Otto Engine Co. builds a solid injection engine of this design.

Price Oil Engine.—It has often been stated as a matter of reproach that oil engines built in the United States have been mere copies or modifications of European designs. This stricture, however, cannot be applied to the Price solid-injection engine.

This engine, the invention of the late Wm. T. Price, was developed in the shops of the De La Vergne Machine Co. while Mr. Price was the chief engineer of that company.

The combustion chamber is pictured in Fig. 330. It will be noticed that the chamber is of a shape such that the oil issuing from the two nozzles in the shape of cones does not strike the iron walls. The section at right angles to the view shown in Fig. 330 appears in Fig. 331. The success of this engine definitely proves that the displacer plug is not necessary to obtain successful combustion if the chamber is so shaped that the oil spray does not strike the cold walls.

By using the two spray nozzles, even if the spray has enough velocity to travel beyond the hot core of air, contact with the iron walls is prevented by collision with the second spray. However, the cooling effect of the walls is greater than in case of the Deutz design, or the hot air forced into the chamber by the Deutz displacer plug adds heat to the air charge, for the Price engine requires a compression pressure of at least 330 lb. to insure ignition upon starting. With a compression of 280 lb. the engine will not fire until repeated compressions warm up the engine. As a means of quick starting an electric coil has been used when the compression was carried at 280 lb. With the Otto or Deutz device compressions of 250 to 260 lb. are ample to give ignition with a 60-deg. suction temperature.

De La Vergne Type S. I. Engine.—The Price combustion chamber is used on the S. I. (solid-injection) engine built by the De La Vergne Machine Co. This company holds shop rights due to the engine being developed in its shops. The S. I. engine is of horizontal design, the units having one, two, three, and four cylinders according to the capacity. The maximum output per cylinder is 180 hp.

The cylinder is cast with the frame, the liner being pressed in. The valve gear is driven by a short shaft in front of the cylinder and by a layshaft along the frame. This layshaft, which carries the governor cam, is driven by gears from the crankshaft.

The governor is of the fly-ball type and controls the amount of fuel passing to the spray nozzle by altering the by-pass setting. The fuel pump is of the simple plunger type, and each pump supplies fuel to the pair of nozzles located in the respective cylinder.

Figure 331 shows a cross-section of a one cylinder unit while Fig. 332 is an exterior view of a twin engine. It will be noticed that the combustion chamber is in two parts. This was done to reduce foundry losses from shrinkage strains and makes the combustion chamber easy to inspect. Replacement cost of either part is also lessened by this design.

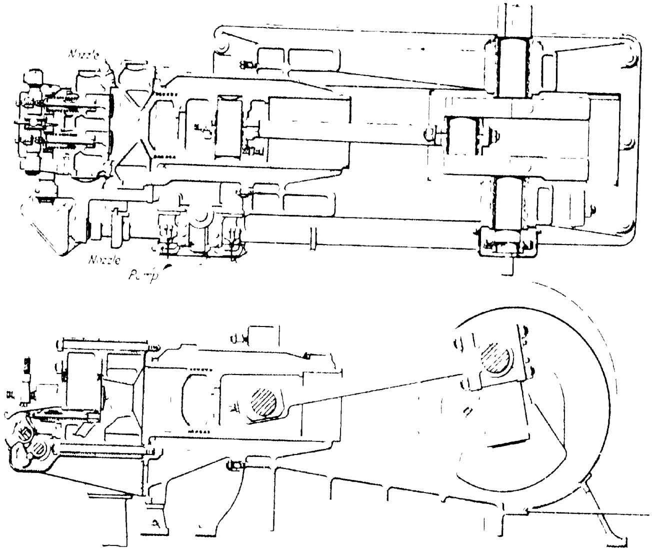


FIG. 331.—Cross-section De La Vergne S I engine

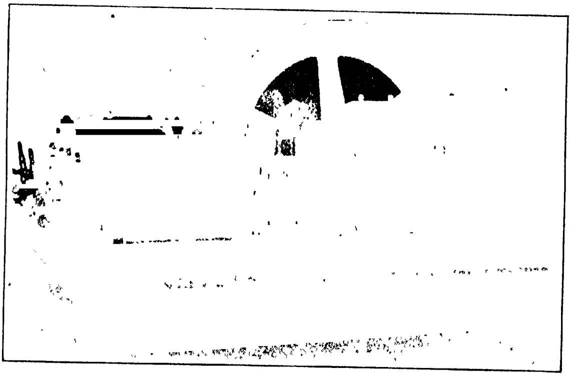


FIG. 332.—De La Vergne two-cylinder engine

The De La Vergne Co. also builds the S. I. engine in vertical units, as illustrated in Fig. 333.

Price-Rathburn Engine.—The Ingersoll-Rand Co. builds the Price engine both in horizontal and vertical units. The first Ingersoll-Rand engines were built at the Rathburn Gas Engine Plant and were called the Price-Rathburn or P-R engines. The combustion chamber is that developed by the late Wm. T. Price. A cross-section of the vertical P-R engine is shown in Fig. 334.

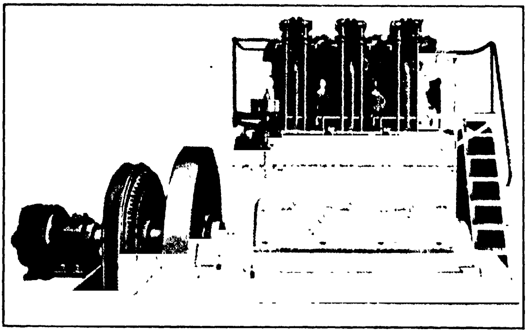


FIG. 333.—De La Vergne vertical S. I. engine.

From a study of this illustration it will be noticed that the frame is of the enclosed box design. The bearings are adjustable—which is a departure from current oil-engine practice. The connecting rod has wedge adjustment at both ends, the piston-pin end being of the slotted type while the crankend has a strap end.

The valves are driven by eccentrics on the layshaft. The valve rocker is unique in that it has a knife-edge fulcrum and the rocker leaves contact at the knife-edge bearing when the eccentrics are at the extreme lower part of their travel. As a result the valves begin to open when the eccentrics are at mid position. At this point the eccentric travel is at a maximum; the valve and closing are rapid.

P-R Horizontal Engine.—The P-R engine is built in horizontal units of 50 and 100 hp. This design is shown in Fig. 335. The cylinder and frame is a one-piece casting with the liner pressed into the jacket. The suction and exhaust valves are driven by cams on a camshaft driven by 2 to 1 spur gears from the crankshaft. The fuel-pump plunger is likewise driven by this

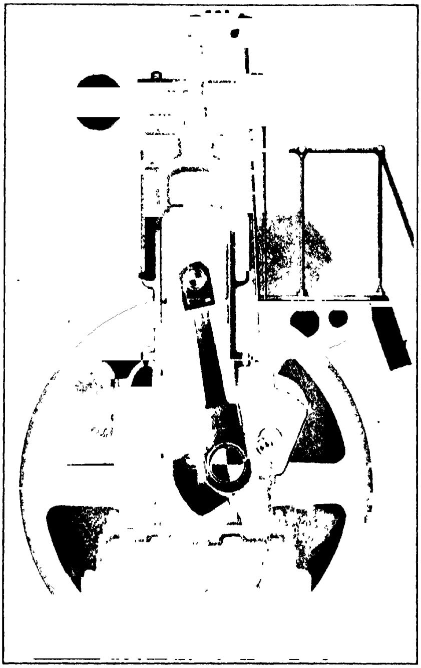


Fig. 334 Ingersoll-Rand vertical Price engine

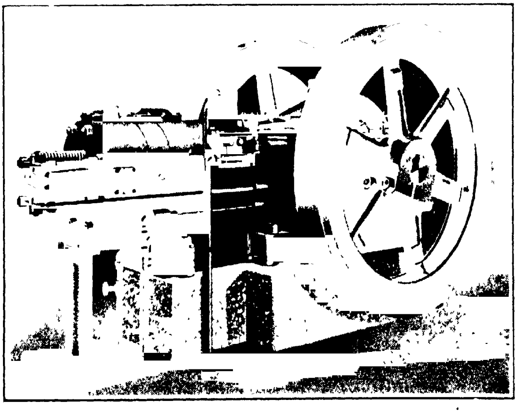


Fig. 335.—Ingersoll-Rand horizontal P-R engine.

camshaft. The governor by shifting a fulcrum block causes the pump plunger to open the by-pass valve earlier or later as needed. This is accomplished by a lever linked to the plunger and contracting with the by-pass valve stem.

The pump is set to start injecting the oil at approximately 6 deg. before dead-center, ending on full load at 6 deg. after dead-center.

Indicator diagrams show that there is an explosion at constant volume, raising the pressure to 500 to 550 lb. followed by combustion at a falling pressure.

Operation of Solid-injection Oil Engine.—As has been outlined heretofore the solid-injection engine operating with 330 lb. compression pressure has with 60 deg. an end-point compression temperature just sufficient to ignite the fuel with a time interval of 0.02 seconds. As soon as the suction air is say 100 deg. F. ignition is positive, even with 280 lb. compression.

It has been found that when the solid-injection engine carries a low load for any length of time the engine cools off so that ignition is irregular with a compression pressure of less than 330 lb. Consequently, after being in service for some time, so that wear causes a loss of compression, the engine behaves irregularly on light loads. To overcome this it is the practice to provide the air intake with a throttle or damper. By wire drawing the suction air through the valve the cylinder pressure on the suction stroke is below atmosphere. There is, then, a reasperation of part of the hot gases in the exhaust line. This hot gas assists in raising the air atmosphere to the igniting point. The same results may be obtained, are obtained to some extent, by causing the valve timing to overlap. Some of the exhaust gases will then be redrawn into the cylinder at all loads. By proper design of the exhaust piping so that the pressure waves in the exhaust pipe at the exhaust valve reach their maximum value before the exhaust valve closes, part of the exhaust gases flow back into the cylinder.

Excessive Pressure.—While the solid-injection engine is not supposed to have a maximum explosive pressure beyond 500 to 550 lb., it is by no means uncommon to find the pressure rising as high as 700 lb. If the engine is built to withstand this high pressure no harm is done. However, such heavy construction means excessive manufacturing costs, and many commercial engines are not so built. It follows that the operator must take steps to eliminate these excessive pressures. In most instances it will be found to be due to poor cooling of the cylinder. This raises the temperature of the suction air which in turn causes the compression-end temperature to be above normal. Ignition then proceeds at a more rapid rate. The operator should see that the cooling-water discharge is increased.

If this does not eliminate the sharp pound, the air suction may be throttled with the idea of so diluting the cylinder charge of air with exhaust gases that the combustion will proceed at a slower rate.

Piston and Rings.—As with the Diesel the piston and rings must be kept clean. When heavy oil is used there is a tendency in the rings to gum fast. This is due to the oil being too viscous to atomize properly or is due to excessive lubricating feed. If the former, the solution is to preheat the oil. Frequently, the deposits occur only upon starting the engine. As the engine is cold at such times, the oil does not burn clean. To overcome this it is advisable to start with kerosene and run on this until the engine warms up. Before shutting down a few minutes' run on kerosene cleans the nozzles and frees the rings of any deposits.

Excessive lubrication oil is of frequent occurrence. Too many engineers are afraid to bring the fuel consumption down to that recommended by the builder.

Fuel Consumption.—The fuel consumption of the solid-injection engine is but little higher than that of the Diesel. While special engines may do better, the fuel consumption of any given series of engines will vary with the compression pressure in about the following rate.

Compression, Lb per Sq. In	Fuel Consumption, Lb per B hp.
250	0 46
300	0 45
350	0 44
400	0 42
450	0 48

Test on Price Engine.—Based on a series of tests conducted at the De La Vergne Machine Co. plant, before a committee of naval officials, on a 19 by 27 engine using the Price injection system, the following report was submitted.

1. The fuel economy under favorable conditions was 0.395 lb. per brake horsepower.

2. Fuel economies of 0.42 lb. at full load could be maintained indefinitely.

3. The engine could be put in operation from a cold condition to full load at full speed in 10 seconds.

4. Any fuel from 15-deg Mexican crude to light fuel oil could be used.

5. At all loads up to 20 per cent overload the exhaust was invisible, and up to 30 per cent overload the exhaust was tolerable.

6. A 24-hr. test, running full load 10 minutes then stopping 5 minutes, proved the engine to be safe against heat stresses, etc.

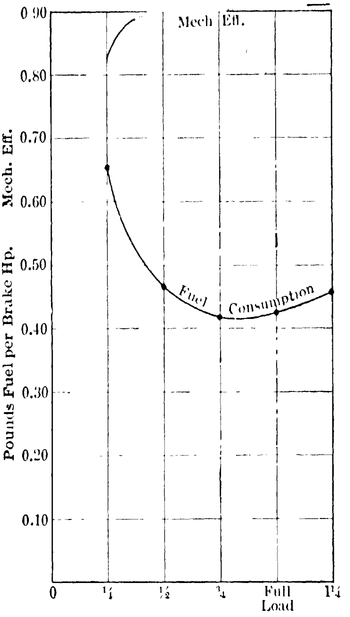


FIG. 336 Test on 19 × 27 Price engine.

7 The engine could be stopped 20 minutes and started again without the assistance of the electrical igniter.

8. A mean pressure of 87 lb. per square inch with respect to brake horsepower, which corresponds to approximately 100 lb. per square inch mean effective pressure, was attained with clean exhaust.

9. The mechanical efficiency when operating above three-quarters load approximated 90 per cent.

Figure 336 presents the results of the test, showing the mechanical efficiency and the fuel per brake horsepower. The fuel consumption compares favorably with the economies obtained from Diesels manufactured in the United States.

Falk Oil Engine.—In Fig. 336A is shown the 550 hp. solid-injection oil engine built by the Falk Corp. This is a four-cylinder unit with a box frame. The cylinder liners are set in a box which rests upon the frame and acts as a cooling-water jacket.

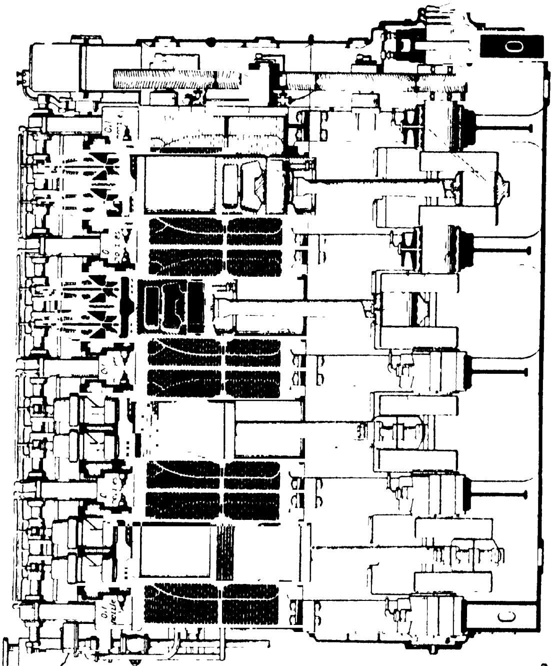


Fig. 336B.—Cross-section of Falk engine.

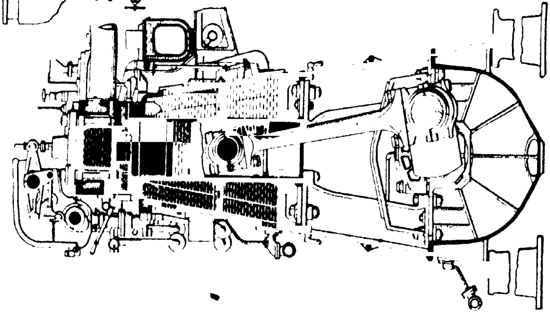


Fig 336A.—Falk 550 hp. solid-injection engine.

It will be observed that the oil is injected through two nozzles into the combustion chamber located in the head. A test on four of these engines geared to a single shaft is given.

SUMMARY OF ENGINE TESTS

Duration of test, hours	3
Load of generator, electric hp	2,073
Efficiency of generator, per cent	92.5
Load on flange on low-speed gear, hp	2,240
Total fuel consumption in three hours, lb	2,851
Fuel consumption per b.hp.-hr., lb	0.424
Speed of low-speed shaft, r.p.m.	88.3
Indicated horsepower, four engines	2,695.4
Mechanical efficiency of engines and gear, per cent	83.1

	Starboard Engines		Port Engines	
	Fore	Aft	Fore	Aft
Exhaust temperature, deg. F	609.7	589.4	629.0	571.6
Cooling water, deg. F	121.0	120.5	120.3	120.4
Indicated hp	669.00	672.00	674.00	680.40

Temperature of:	
Cooling water at inlet, deg. F	38.5
Fuel oil in measuring tanks, deg. F	88.1
Lubricating oil at inlet, deg. F.	98.0
Lubricating oil at tank, deg. F	103.0
Pressure of:	
Lubricating oil at pump, lb	15.8
Lubricating oil at engines, lb	8.1

Lubricating Oil Consumption.—The amount of lubricating oil used by the solid-injection engine (this includes the gas-injection or precombustion engine) is not as uniform regardless of the make as in case of the Diesel. It is by no means unusual to find the oil varying from 1 gal. per 750 hp.-hr. to 1 gal. per 2,000 hp.-hr. This is a very important item for purchasers to consider, and the builder should be called upon for guaranteed figures.

CHAPTER XXI

INDICATING THE DIESEL ENGINE

General.—The value of an indicator in locating engine faults is not understood by the average Diesel engineer. For this reason the process of indicating the engine will be covered in detail.

If a sheet of cross-section paper were laid off so that horizontal measurements represented to some scale the distance traveled by the piston of an engine from the beginning of the stroke, and vertical measurements represented to scale the pressure existing within the cylinder at the various points in the stroke, a diagram similar to Fig. 337 would be developed. Here a charge of pure air is drawn into the cylinder as the piston moves in its suction stroke. Since the pressure is constant while the volume increases from the clearance volume indicated by the point *F* to *A* (*FA* being equal to the stroke volume, or the piston stroke in feet times the piston area in square feet), the line *FA* would represent the volume and pressure conditions. At the end of the suction stroke the piston reverses and compresses the charge of air, the pressure and volume changes following the compression curve *AB*.

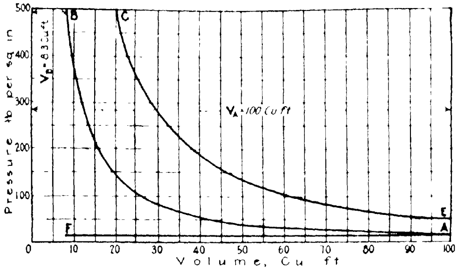


FIG. 337.—Diagram showing cylinder pressure at all points in the piston stroke.

Just as the piston reaches the end of the compression stroke, the fuel valve opens and allows the fuel charge to blow into the combustion space. The piston, under the action of the pressure generated by the burning oil, is forced out on the working stroke. The rate of oil injection is such that the pressure during the combustion period remains constant. This is shown by the line *BC*. At *C* the fuel valve closes and the hot gases in the cylinder expand behind the moving piston, giving the expansion curve *CE*. At *E*, the end of the stroke, the exhaust valve opens and the gases flow out through the exhaust passage, causing the cylinder pressure to drop to that of the atmosphere, this being represented by the line *EA*. The gas remaining in the cylinder is forced out by the piston on the exhaust stroke, tracing the line *AF* coinciding with the suction line *FA*.

The several events on the diagram may be listed as follows:

Suction Line.—Fresh air is drawn into the cylinder by the suction of the piston, filling the space vacated by the retreating piston; line *FA*.

Compression Line.—The reversal of the piston and its movement toward the cylinder head compresses the air trapped in the cylinder until at the end of the stroke, the compression pressure is around 48 to 500 lb. gage; line *AB*.

Combustion.—The fuel or spray valve opens, allowing the high-pressure injection air to blow the fuel charge into the cylinder. The rate of fuel admission is such that no increase in pressure occurs, the heat added being sufficient to keep the pressure practically constant; line *BC*.

Expansion.—The fuel valve closes and, the combustion being complete, the hot gases expand forcing the piston to the end of the stroke; line *CE*.

Exhaust.—At or near the end of the stroke the exhaust valve opens, the hot gases flow out of the cylinder; line *EA*. The cylinder pressure drops to atmospheric and on the next stroke the piston forces the remainder of the gases out of the cylinder; line *AF*.

If we could have such a diagram, giving the pressure in the cylinder at every point it would show the freedom of the intake, the work lost by suction, the extent and perfection of the compression, the point at which ignition takes place, the pressure realized by combustion or explosion and the rapidity of the process, and the manner in which the heated gases expand; also the power that the engine is developing, as well as defects in the mechanical details of the engine.

The Indicator.—James Watt, who is regarded as the originator of the practical steam engine, set about to construct some contrivance to obtain such a diagram of the cylinder events. The result of his labor was the Watt steam-engine indicator shown in Fig. 338. Here the indicator is attached to the cylinder and the diagram board *V* which slides in the frame *U* is connected by a cord *O* to a rocker arm attached to the crosshead; the weight *W* holds the cord taut. Since the indicator cylinder *S*, which contains a piston *T* and a spring *H*, is connected to the steam cylinder, the steam exerts the same pressure per square inch on the piston *T* as it exerts on the engine piston and overcomes the spring resistance. The pencil *P* at the end of the indicator piston rod then rises and falls with the changes in pressure within the engine cylinder.

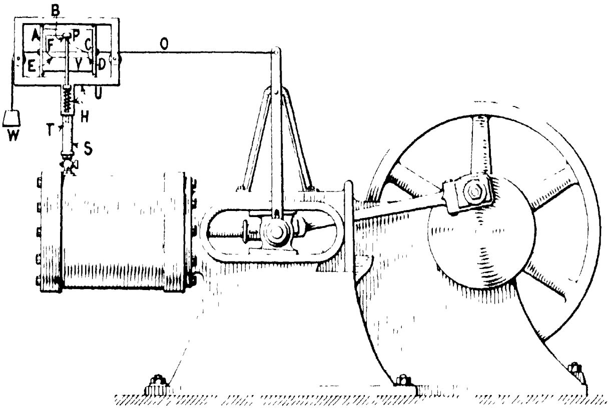


FIG. 338. Watt's engine indicator

As all know who have used spring balances, the heavier the weight loaded on the scale pan the greater is the movement of the spring, and the scale reading, although marked in pounds, actually represents the movement of the spring. The motion of the piston *T* is transmitted to the pencil *P*. If the engine piston is at the end of the stroke and boiler pressure is turned on, the pencil will be at the point *A*. As the piston moves on, the stroke the board travels with the crosshead and the pencil makes a line *AB*, where cutoff takes place, whereupon the steam expands along *BC*. The opening of the exhaust valve reduces the pressure on the piston, and the pencil moves to *D*. On the return stroke the weight holds the cords tight and the line *DE* is made by the pencil. At *E* compression takes place and the line *EF* is made. The atmospheric line is made by cutting off the opening to the cylinder and allowing the indicator cylinder to be open to the atmosphere, whereupon the pressure on the bottom of the indicator piston being that due to the atmosphere, a straight line is obtained at a distance from the zero pressure line proportional to the pressure of the atmosphere. The reader will observe that at any point in the piston stroke the pressure within the engine cylinder is the same as that in the indicator cylinder, and the pencil position is proportional to this pressure. The diagram so secured is then a representation to scale of the events in the engine cylinder.

While this indicator, with all its faults of great weight, etc., was fairly satisfactory for engines making 30 strokes or less per minute, it was of little use when engine speeds were increased, since the length of spring and weight of the piston were such that at the instant steam was admitted to the engine, the inertia of the indicator parts tended to cause the indicator piston to travel higher than it should. The Watt indicator was improved and developed with the idea of reducing the inertia and errors; the first step was to convert the diagram board into a drum. One of these early indicators, that of McNaught, where the paper was placed on a light drum which was rotated by a cord from the engine crosshead. The indicator did not become a serviceable instrument until Charles B. Richards, of Providence, R. I., brought out the Richards indicator. This instrument was fitted with a strong spring giving a very small travel to the piston, which eliminated the inertia of the long springs used before.

Modern Indicators.—Modern indicators followed in general the design brought out by Richards and until the last few years were all made with an inside spring, as shown in Fig. 339. Here the working barrel *A* is surrounded by the housing *C*, the space between the two acting as a gas jacket, with the idea of avoiding distortions of the barrel from uneven temperatures. The lower end of the barrel is free to accommodate itself to change due to varying steam temperatures. The piston *B* works inside the barrel and is provided with a spring, the resistance of which causes the piston travel to be proportionate to the gas pressure under the piston. The upper end of the spring is attached to the indicator cap. The piston is provided with a piston rod which carries at its upper end the swivel head *F* to which is attached the pencil linkage as shown. The drum, about which is wrapped the paper card, is carried on an arm projecting from the indicator housing. The cord *G* is attached to the engine reducing motion, and its pull is opposed by the tension of the drum spring. This arrangement insures the cord being held taut at all times, and when the cord on the out stroke of the engine rotates the drum, the spring tension brings the drum back to its original position on the return-stroke of the engine.

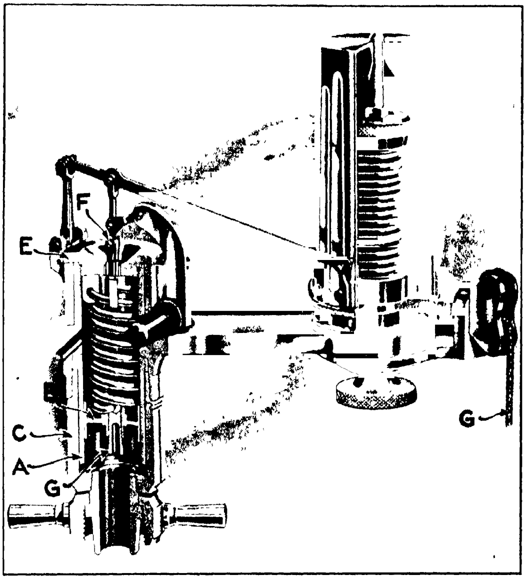


Fig. 339.—Typical inside spring indicator. (Crosby).

As long as pressures and temperatures were not high, the inside type of indicator proved to be perfectly satisfactory. Under modern gas- and oil-engine conditions high temperatures altered the characteristics of the spring, giving unreliable diagrams. The spring was placed on top of the indicator, away from danger of excessive temperature changes, and the majority of instruments manufactured in recent years are of this type, see Fig. 340.

Owing to the high pressures in the heavy oil engine the steam-engine-indicator piston, which has $\frac{1}{2}$-sq. in. area, would require an extremely heavy indicator spring. It is usual to fit the oil-engine indicator with a piston of $\frac{1}{4}$-sq. in. area. In this way the 200-lb. steam-engine spring may be used and will then allow

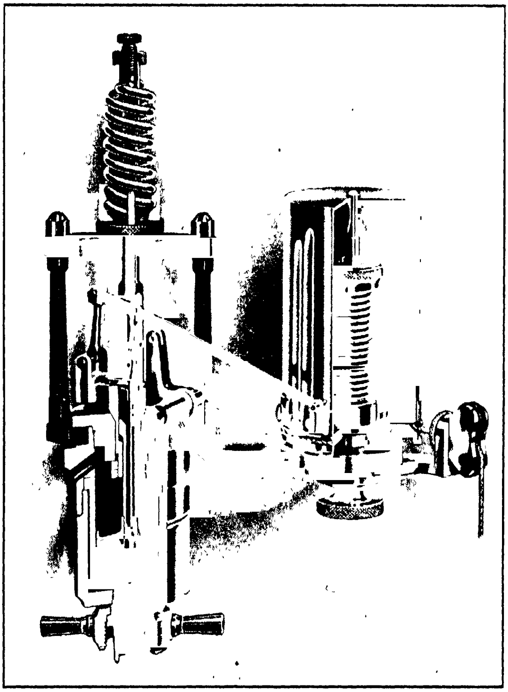


Fig. 340.—Crosby outside spring indicator.

1 in. of the travel of the pencil when a pressure of 400 lb. per square inch is exerted in the engine cylinder.

Hopkinson Optical Indicator.—To permit securing diagrams from engines operating at high speed, various optical indicators have been designed. The Hopkinson indicator, Fig. 341, has been more largely used than any other of this class. A barrel *A* contains a piston *F* which is held by a shackle *G* to the flat spring *D*. The spring is supported upon two arms by the screws *E*.

A mirror *H* is pivoted to the shaft *I*, which is connected by the arm *M* and the stirrup *L* to the spring. A slight deflection of the spring causes the mirror to rotate a slight amount on its axis. If a ray of light is allowed to strike the mirror, the partial rotation of the mirror will cause the image of the light to move in phase with the indicator piston, but the ratio of travel is large. The mirror and spring are mounted on a collar *B* which is rotated by the indicator cord. The two motions of the mirror cause the image of light to trace on a plane surface a diagram representing the pressure-volume conditions within the engine cylinder. If a sensitive plate is used, a photographic diagram results.

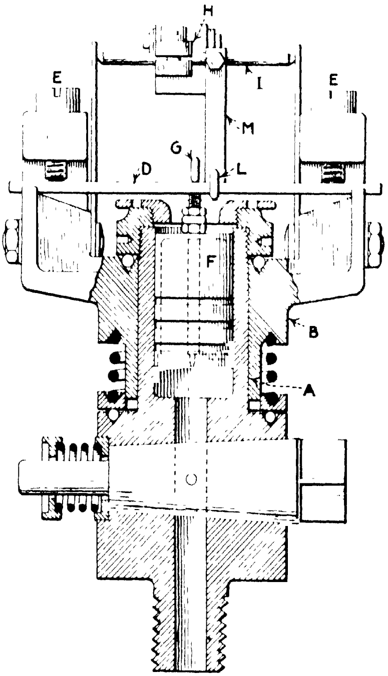


Fig. 341.—Hopkinson optical indicator

Midgley Indicator.—A recent optical indicator designed for gas-engine work, but equally suitable for steam-engine testing, is the Midgley, illustrated in Fig. 342. In this instrument the indicator proper is fitted with a piston and a special spring. The piston rod is linked to a mirror at its top. Light rays from an electric bulb are thrown upon this mirror, which is shifted on a horizontal axis by the movement of the indicator piston. The rays are then reflected back upon an eight-sided mirror and thence upon the ground-glass screen, rising and falling with the change in pressure. The eight-sided mirror is driven from the engine in phase with the engine piston. This gives a horizontal movement to the ray of light. A diagram showing the cylinder events is

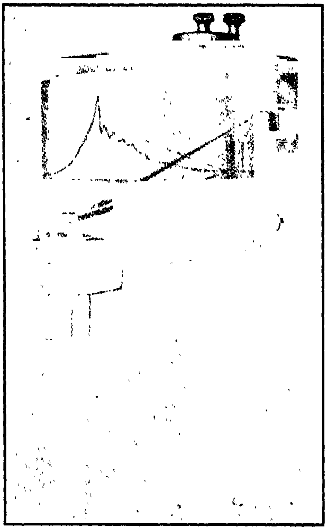


FIG. 342.—Midgley highspeed optical indicator.

then made on a sensitive plate or is observed by the reflection on the ground glass.

Comparisons of Optical and Standard Indicators.—While early indicators often gave distorted diagrams, present-day instruments are apparently as accurate for speeds up to 600 r.p.m. as are optical indicators. A number of years ago several tests, as reported in *Engineering*, were made on the Hopkinson and piston indicators. These tests apparently proved that the optical device was no more accurate than the standard instrument. Beyond a speed of 600 r.p.m. the inertia of the indicator parts is such that the diagram secured is by no means accurate, and little dependence should be placed upon the diagram from a piston-type indicator. On the other hand, the optical instrument gives fairly reliable diagrams up to 1,000 r.p.m.

The pressure scale has so large a ratio that the usual oil-engine indicator diagram may not be used to measure the indicated horsepower of the engine with the same degree of accuracy as in case of the steam engine having lower pressures. For computations where an error of 5 per cent is allowable, the indicated horsepower from the diagram is quite satisfactory. For accurate study of cylinder events the optical instrument is the only satisfactory solution.

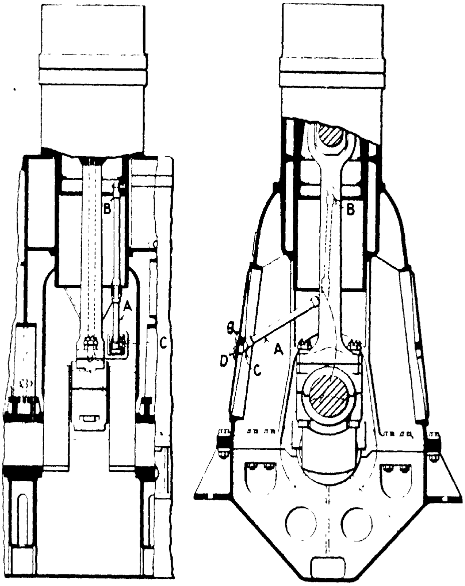


FIG 343.—Indicator rigging for trunk piston engines.

Indicator Rigs.—The majority of Diesel engines of 700 hp. and under have trunk pistons. For this reason the indicator riggings usually employed on steam engines may not be used on the Diesel. Even the large crosshead-type engine requires some special rigging. In Fig. 343 is shown a type of rigging often used on trunk pistons. The lever *B* is fastened to the inner wall of the piston by a screwed pin. The rocker *A* is fulcrumed on a pin *C* at the side of the crankcase door and is fastened to lever *B* by a bolt and knuckle joint. The indicator cord is looped over the hooked end *D* of the rocker *A*. There is a slight error in the diagram resulting from this rigging. If, however, the lever *A* is made of such a length that the knuckle joining *A* to *B* moves equal distances on the two sides of the cylinder center line, the error is negligible. This rigging is easy to make if the engine builder has not supplied some arrangement.

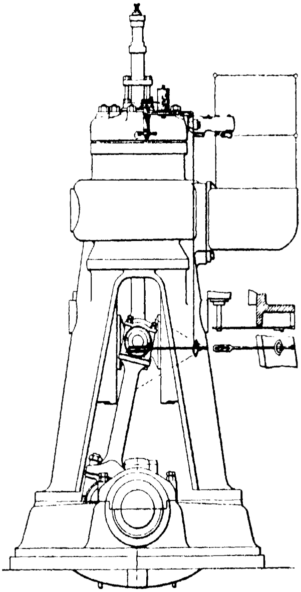


FIG. 344.—Rigging used on crosshead-type engines.

With this rigging, when the piston moves upward the indicator-drum spring tension is increased. If the cord has a tendency to stretch, the diagrams may be distorted toward the end of the compression stroke and during the combustion period. Errors at this point are of serious moment since the most important events in the engine cycle take place along this portion of the diagram.

Figure 344 outlines an arrangement that is quite satisfactory for crosshead-type engines. By reason of the distance the crosshead is from the face of the frame, it is seldom possible to link the rocker arm direct to the crosshead. Usually, a stud of some length carrying the lever at the outer end must be screwed into the crosshead.

It will be noticed that the diagrams are accurate if the quadrant over which the cord passes is correctly made. The cord connection is such that, as the piston moves downward, the indicator cord is drawn out against the spring tension. If the cord stretches, distortions of the diagram will occur toward the end of the expansion stroke, but this is not as serious as an equal amount of stretching at the other end of the diagram.

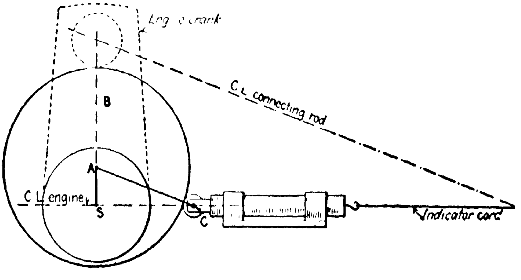


FIG. 345. Easily constructed rigging

Many of the engines now in use were not fitted with indicator rigging and the engineer must supply the deficiency. Either of the two arrangements discussed may be used. Simpler and more easily applied devices are often made by the engineer. In Fig. 345 appears a device that is easily made and gives a correct diagram. This consists of an eccentric B clamped to the engine shaft with the eccentric throw SA in line with the engine crank. In constructing this motion, the relation of the radius of the eccentric B plus the radius of the roller C (the distance AC) to the engine connecting rod is equal to the ratio of the eccentric throw to the engine-crank throw, or

$$\frac{AC}{\text{Connecting rod}} = \frac{AS}{\text{Crank throw}}$$

Any engineer can make this, using a piece of steel rod for the slide and a piece of pipe with a babbitt liner for the guides. The eccentric can be made of cast iron, split and drilled for clamp bolts.

In an emergency a hardwood eccentric with a brass strip on the edge may be used.

Instead of an eccentric the end of the engine shaft can be drilled for a small capscrew. A connecting rod of steel bar of a length such that the distance from the engine shaft *S* to the capscrew *A* bears the same ratio to the crank throw that the engine connecting rod bears to the rod *AC* will give a miniature connecting rod *AC* and crank *SA* that reproduce to a smaller scale the action of the engine crank and rod.

An eccentric may be placed on the engine crankshaft and a lever *A* with a groove placed in the groove and fulcrumed to a stand *B*, Fig. 346. If the line connecting the center of the eccentric and shaft coincides with the crank line, then the travel of any point on the rod *A* will be proportional to the piston motion at all points in the crank circle. Hooking the indicator cord to a point on the lever such that the cord travel gives the desired diagram length, the apparatus is ready for use and the engine may be indicated.

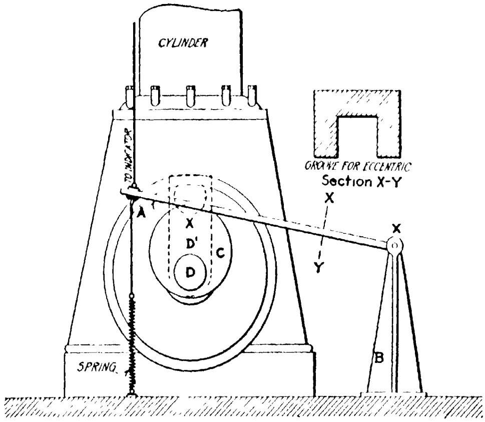


FIG. 346.—Rigging using an eccentric.

Figure 347 outlines the rig employed by the Worthington Pump and Machinery Corp. on the Cambridge two-stroke-cycle solid-injection engine. A link pinned to the engine crosshead transmits the piston motion to the lever *A*, which, through a short arm *B*, a vertical rod *C*, a bell crank *D*, and a shaft *E*, pulls the indicator drum cord. This in appearance is very neat, but after a short time if the parts are not kept snug, the slight wear at the joints causes the indicator diagram to be unreliable.

Indicator Cocks.—The passage through the engine cylinder head to the indicator cock is usually very small, and if a stop cock is placed between cylinder and indicator cock this passage through the stop cock is still smaller. It may happen that these passages will become partially choked with carbonized oil, especially in the stop cock. If this proceeds far enough, the passage will become constricted enough to reduce the gas flow sufficiently to give the effect of a lower pressure. As a result the operator may be misled as to the existing compression.

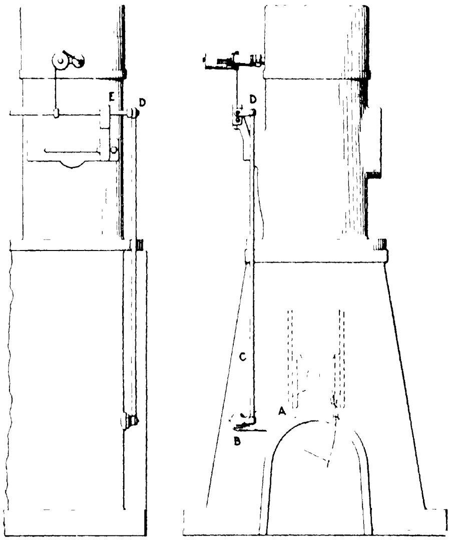


FIG. 317.—Rigging on Worthington Diesel.

Anyone who has taken diagrams from an oil engine has had burned fingers as evidence of the temperature acquired by the whole indicator after a few minutes' use. Aside from the difficulty of handling a hot indicator consideration must be given to the bad effect on the indicator piston and cylinder. If the indicator remains on the engine for any considerable length of time and a large number of consecutive diagrams are taken, as in testing floor work, the piston will soon become leaky. A very useful device to prevent this and to eliminate from indicator work the burned-finger annoyance is a water-cooled stop cock, shown in Fig. 348. This can be left permanently attached to the engine cylinder, and the water-jacketed inlet and outlet may be connected at any convenient point in the circulating water system by means of small tubing. An indicator may remain attached to this cock for an indefinite period and will not get hot as long as the cooling water is circulating.

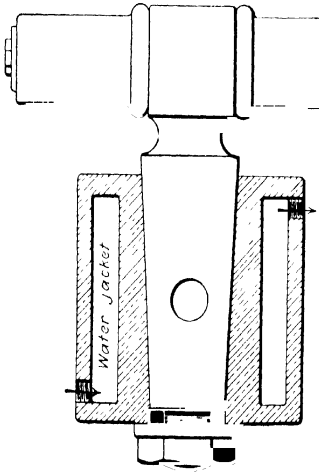


Fig. 348. Water-cooled indicator cock

The Air-cycle Diagram.—In the construction of Fig. 337 the assumption is made that nothing but pure air exists in its cylinder and that the heat is added to the air, there being no gaseous products of combustion. It is further assumed that no heat transfer takes place between the air in the cylinder and the cylinder walls; that is, both expansion and compression are adiabatic. The exponent in the equation for adiabatic expansion and compression $P_1V_1^n = P_2V_2^n$ is taken as 1.408 for air. The reader is referred to Chapter XX for a discussion on this question.

Actual Diesel Diagram.—This ideal diagram is never obtained from a working engine for several reasons. In the first place the cylinder charge of air loses more or less heat to the cylinder walls during the compression period. This causes the compression curve to fall below the adiabatic curve for air. In addition, during the expansion stroke the hot cylinder walls give off heat to the expanding gases. The actual expansion line is above the theoretical expansion line on the air cycle diagram, Fig. 337. The specific heat of the gases does not remain constant during the expansion stroke, and the fuel is not completely burned when the fuel valve closes at C, but continues to burn through a considerable portion of the expansion stroke. This also changes the slope of the expansion curve.

It is generally assumed that the exponent of the compression and expansion equation PV^n = constant is 1.35 instead of 1.408 for air. As a result of the heat loss during compression the final clearance volume of the Diesel cylinder must be less than the air-cycle clearance to attain the same final pressure. If it be assumed that the amount of air charge in Fig. 337 is just sufficient for the combustion of a fuel charge that will hold the pressure constant to C, then the point C in the actual engine will fall closer to the clearance line. The diagram of the actual engine will then be similar to the full-line diagram in Fig. 349, the ideal diagram being shown in broken lines. The expansion in the working engine is not carried to the end of the stroke, but release occurs at E^1, this being when the crank is at an angle of 30 to 50 deg. from dead center, the piston having completed from 80 to 90 per cent of its stroke. The exhaust and suction lines are not shown as the departure from the ideal line FA will be but slight.

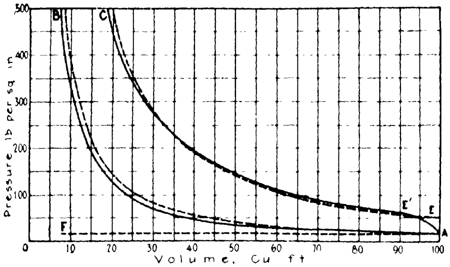


FIG. 349.- Diagram showing departure from air-cycle diagram.

In addition it will be found that the diagram taken from the working engine will show other departures from the ideal diagram. The cylinder pressure during the combustion period BC is never constant; the line BC then departs somewhat from the horizontal.

It is possible, however, by careful regulation of the injection air pressure and the number of atomizer disks in the fuel valve to maintain constant pressure for a portion of the combustion period as illustrated in Fig. 350. In Fig. 351 is shown a diagram

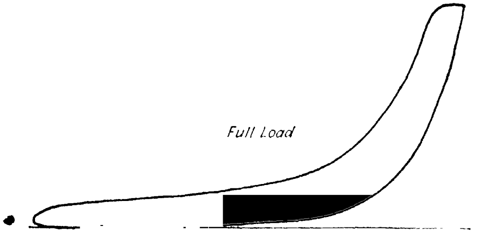


FIG. 350.—Actual Diesel indicator diagram.

where the pressure during combustion shows an increase. This may be traced to the few atomizing disks in the fuel valve

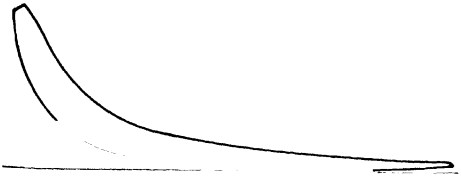


FIG. 351.—Diagram showing high injection-air pressure.

together with a high injection pressure. To correct this condition the air pressure should have been decreased and one or two

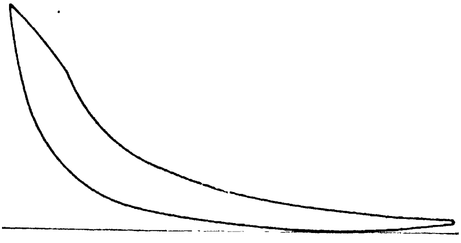


FIG. 352.—Usual form of combustion line.

more atomizer disks added, or a few of the holes through the present disks closed. This will increase the resistance to the oil flow and so decrease the rate of oil injection.

Variations in the Combustion Line.—Often, the Diesel diagram has a combustion line somewhat as in Fig. 352, the pressures falling during the period of fuel injection and combustion. It has been found that the action of the engine is improved, the efficiency is not affected and there can be a wider variation of air injection pressure, oil viscosity, etc., without causing trouble.

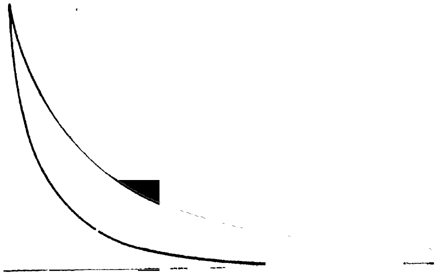


Fig. 353.—Diagram shows late fuel injection.

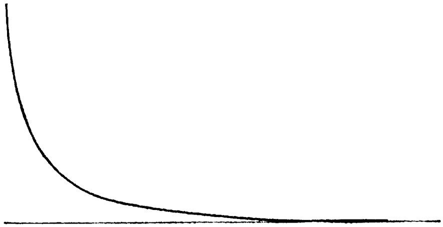


Fig. 354. Compression diagram.

If the fuel injection is late, the shape of the combustion line becomes similar to Fig. 353, giving the appearance of a sharp peak to the diagram. If injection is early, the same character of peaked diagram is obtained and it is often quite impossible

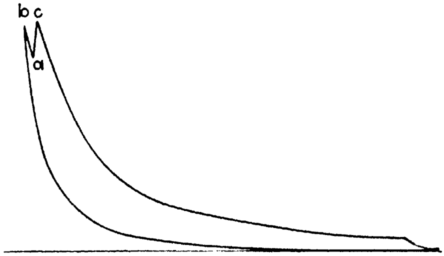


Fig. 355.—Combustion extremely slow.

to tell by inspection whether the fuel cam is set to give early or late fuel injection. The only way to locate the trouble is to cut out the fuel to the cylinder in question and take a compression diagram, Fig. 354. This diagram shows the compression and re-expansion of the cylinder charge of air. The pressure at the top of the curve is the total compression pressure. Comparing the height of this diagram with that of the diagram taken showing the peak, will reveal whether the pressure at the peak is higher than the compression pressure. If it is higher, then the fuel admission is early. If no increase in pressure is found, the fuel injection must have been late to cause the peak.

Late injection does not always cause a sloping combustion line. At times the diagram appears as shown in Fig. 355 which was obtained from a two-stroke-cycle Diesel. The cylinder pressure upon the reversal of the piston dropped from *b* to *a* by reason of the slow rate of combustion at which point the increased ignition of the fuel raised the pressure to *c*, practically as high as at *b*.

Suction and Exhaust Events.—Since the spring used in the indicator is at least heavy enough to require 200 lb. per square inch to give a pencil travel of 1 in., the suction and exhaust lines cannot be studied to any advantage. It is frequently desirable to examine these lines to detect any restriction due to long pipe lines or to improper valve setting. If a light spring is used in the indicator, so that the small variations in the suction and exhaust pressures are shown, the high pressures during combustion would cause the light spring to be jammed, probably breaking it. To avoid this, a light brass tube *C* may be slipped over the piston rod before the spring is screwed on, as shown in Fig. 356. The length of the tube should be about one-half the length of the spring.

In Fig. 356 at *A* is shown an indicator diagram obtained with this arrangement. Here the suction pressure is shown to be several pounds below the atmosphere line, while the exhaust is probably less than 1 lb. above atmosphere. The compression line rises until the spring end touches the brass tube, whereupon no higher pressures are shown, the pencil making a horizontal line to the end of the compression stroke. On the combustion and expansion stroke, the pressure conditions are not shown until the pressure drops to that indicated by the spring when the brass tube stopped the spring compression. At this point the opening of the exhaust valve caused the cylinder pressure to drop sharply as shown at release. Only in installations where the exhaust or suction lines are long or where the exhausts of several cylinders interfere with one another will the pressures during this part of the cycle differ much from atmospheric; a drop of more than 1 lb. should not be allowed to exist.

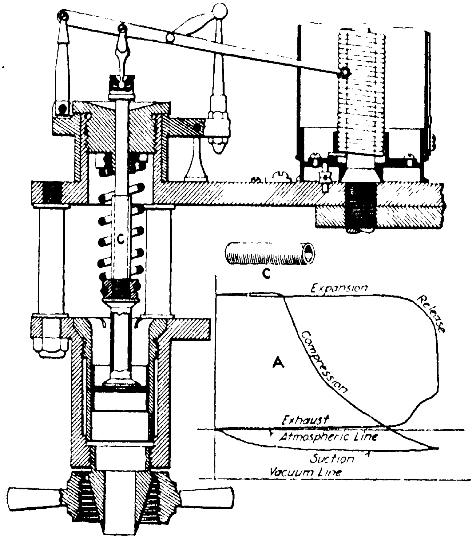


FIG. 356. Indicator with a weak spring

Distorted Diagrams.—It will be noticed that the combustion events are crowded together at one end of the diagram, although during this time the crank passes over a considerable percentage of its circle, about 40 deg. If this end portion of the diagram could be lengthened, it could be studied with better results. To do this, it is customary to connect the indicator rigging so that instead of the indicator drum travel being in phase with the engine piston, it is out of phase. This means that when the engine piston is near the end of the stroke and traveling at its slowest rate, the indicator drum is at the center of its travel and traveling at its highest speed. To secure an out-of-phase diagram, the indicator cord can be attached to another engine piston or cross head in place of the piston of the cylinder being indicated.

It is not absolutely necessary to so connect the indicator cord. Instead, the cord may be pulled by hand, for all that is desired is a picture of the combustion events. After a few trials the engineer can easily pull the drum cord rapidly when the piston is firing, even though the speed be as high as 300 r.p.m. The diagram thus obtained, if no fuel is being injected, will be similar to Fig. 357. In the ordinary diagram there would be a sharp peak at the end of compression; with this out-of-phase diagram the top is rounded.

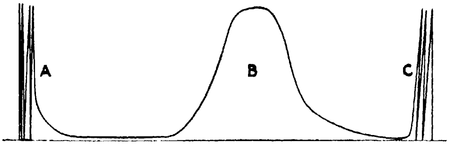


Fig. 357.—Indicator drum cord pulled by hand.

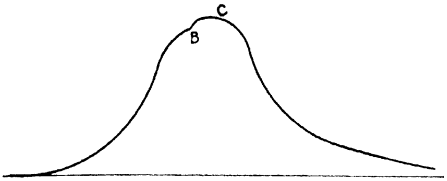


Fig. 358.—Diagram shows normal injection.

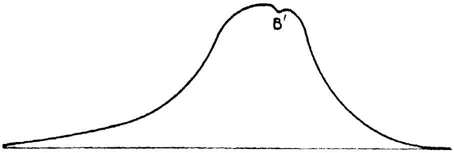


Fig. 359.—Fuel injection is late.

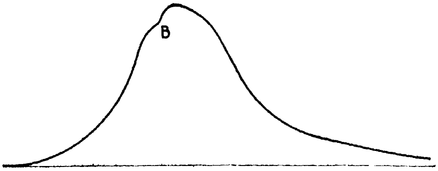


Fig. 360.—Effect of early ignition.

When the engine is firing, the rounded top becomes wider and variation on pressure is made apparent. In Fig. 358 the fuel was ignited at *B* with the piston at dead center and continued to *C*. This is the normal diagram but in Fig. 359 the fuel did not ignite until B^1, showing late fuel ignition, while in Fig. 360 the fuel ignited early at *B*. These distorted diagrams will reveal combustion defects and may be taken by connecting the indicator to the cylinder head without the necessity of shutting down to hook up an indicator rigging.

McEwen Diesel Indicator Diagrams.—Figure 361 shows diagrams from a 14 by 22 McEwen Diesel. All show a slight drop in the admission line, indicating that the resistance in the nozzle was too great for the injection-air pressure carried. Raising the air-blast pressure would tend to bring the admission line to a horizontal position. For the sake of clearness the exhaust and suction strokes are not indicated.

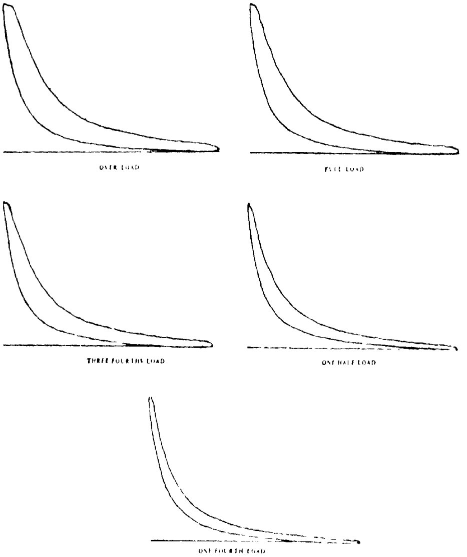


FIG. 361.—Indicator diagrams, McEwen engine.

Allis-Chalmers Indicator Diagrams.—Figure 362 shows typical diagrams from the Allis-Chalmers open-nozzle Diesel. The rising slope of the admission or combustion line indicates that the fuel, although injection began early, did not ignite readily at

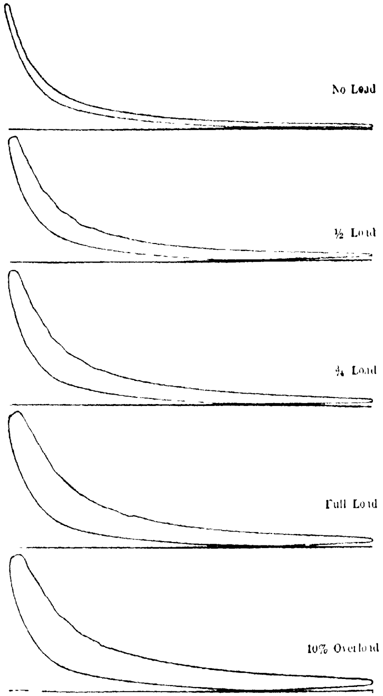


FIG. 362.—Indicator diagrams, Allis-Chalmers Diesel

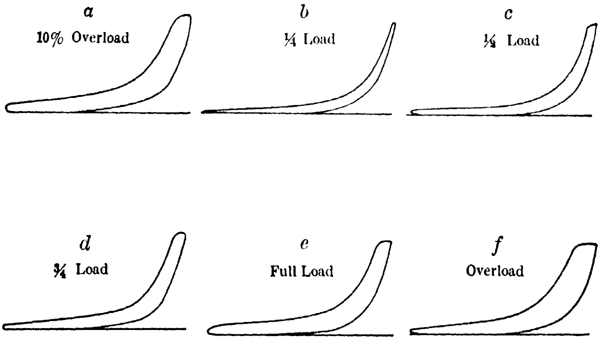

a 10% Overload
b ¼ Load
c ½ Load
d ¾ Load
e Full Load
f Overload

FIG. 363.—Indicator diagrams, National Transit Diesel.

the beginning of injection only the lighter portions ignited. This raised the temperature sufficiently to ignite the entire charge. The effect was accumulative, producing the rising line.

National Transit Indicator Diagrams.—Figure 363 depicts diagrams from a 15½ by 24 National Transit Diesel engine at 180 r.p.m. The notable feature of these diagrams is the horizontal combustion line, produced, evidently, by a happy combination of efficient atomization and proper air-blast pressure. The quarter-load card shows a sharp peak at the end of the compression. Some attribute this to inertia in the indicator. It is very probable that this peak is a result of the inability of the air blast to pick up the fuel at the instant of valve opening. The air had to attain a high velocity before the oil was swept into the cylinder. The piston, in the meantime, had retreated; the combustion of the fuel was insufficient to keep the pressure constant.

Standard Fuel Oil Indicator Diagrams.—Figure 364 shows diagrams for the Standard Fuel Oil two-stroke-cycle Diesel engine, and they are typical cards from an engine working under this cycle.

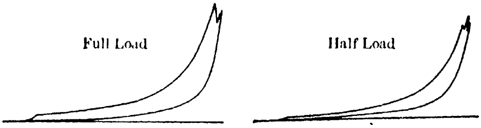


FIG. 364.—Indicator diagrams, Standard two-stroke-cycle engine.

Errors to Avoid in Indicating a Diesel Engine.—Manual facility in using an indicator is quickly acquired by practice, but it is often the case that oil-engine indicator diagrams are not correctly interpreted because the operator does not have a full appreciation of the many mechanical factors that affect the operation of the instrument.

A prime requisite in an indicator for oil-engine work is heavy and rigid construction. This applies to both the indicator and its attachments, such as stop cocks, elbows, etc. Such a design is necessary not only to enable the indicator to withstand the ordinary wear and tear in service, but also to give the indicator assembly rigidity enough to prevent deflection due to the pull of the cord against the spring.

A simple method of testing the indicator for rigidity is to put a light wooden stick as a brace between the drum arm and the engine cylinder, as shown in Fig. 365, take a card, then remove the brace and trace another diagram on the same card. Figure 366 is a reproduction of two actual diagrams taken in this manner.

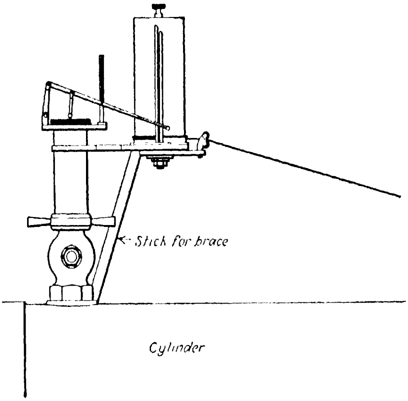


FIG. 365.—Making the indicator rigid.

The full-line diagram was traced with the brace in position, and the dotted-line diagram was taken with brace removed. This shows that the diagram was shortened about $\frac{1}{16}$ in. and the combustion line distorted by bending of the drum arm on the indicator, and illustrates very clearly the necessity for rigid construction.

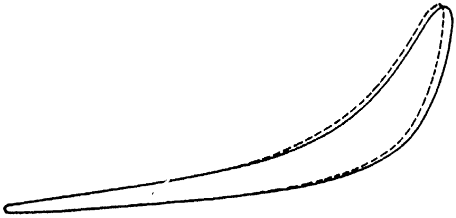


FIG. 366.—Effect of weak indicator.

Stretch of Cord.—Another common cause of error in indicator diagrams is the stretch of indicator cord. On this account the use of any kind of cord should be avoided wherever possible and the connection between indicator and reducing motion should be made by means of fine wire or steel ribbon. The majority of vertical oil engines have a lever-reducing motion so that when the piston is at the top of its stroke the cord is pulled out against the drum-spring tension. An example of the effect of a stretching cord is shown in Fig. 367. This diagram might be diagnosed as a case of retarded injection, incorrect valve setting or slow burning, but Fig. 368, which is a diagram taken immediately afterward from the same cylinder using a wire in place of the cord, shows the trouble to have been caused by stretching of the cord. In this case this conclusion was verified by pulling the cord by hand, and it was found possible to stretch it about 2 in. Study of the diagram will show that at the beginning of the compression stroke a normal diagram was traced, but toward the end of the compression stroke the drum-spring tension became sufficient to start the cord stretching, and the drum then remained stationary until the rest of the compression stroke was completed and enough of the working stroke to take up the stretch of the cord.

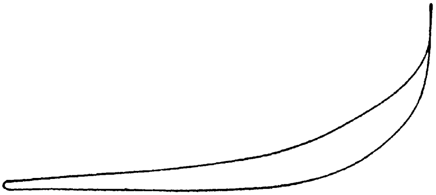


FIG. 367.—Stretched drum cord gave this diagram.

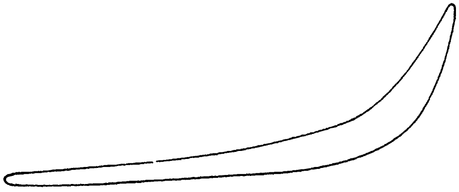


FIG. 368.—Wire cord caused this diagram.

Indicator Passage.—The passage through the engine cylinder head to the indicator cock is usually very small, and if a stop cock is placed between cylinder and indicator cock, the passage through the stop cock is still smaller. It may happen that these passages will become partly choked with carbonized oil, especially in the stop cock. If this proceeds far enough the passage will become constricted enough to reduce the gas flow sufficiently to give the effect of a very much "leaner" diagram. Figure 369 shows in heavy lines a diagram taken under this condition. This diagram is apparently normal, but a knowledge of the approximate power the cylinder had been developing at the time the diagram was taken indicated that the diagram was too small. The diagram shown in dotted lines and superimposed on the first one was taken after the stop cock had been cleaned out.

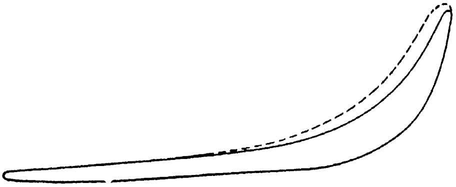


Fig. 369.—Effect of restricted gas passage

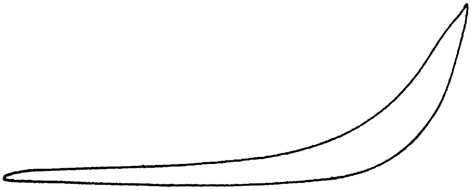


Fig 370 —Indicator piston leaked.

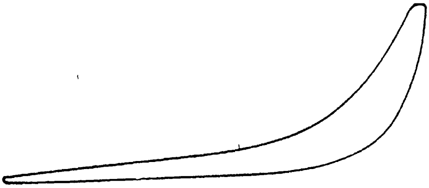


Fig. 371.—Correct diagram with good indicator.

Leaky Indicator Piston.—The effect of slight piston leakage is shown in a recent test on a large Diesel engine. A diagram, Fig. 370, was taken with a rather light indicator having a somewhat worn piston, and immediately afterwards the diagram in Fig. 371 was taken from the same cylinder under exactly the same conditions with an exceptionally fine, strongly built instrument. When measured, the second diagram gave about 7 lb. more mean effective pressure than did the first one. This test was repeated several times to prove that the apparent variation in mean effective pressure was due to difference in indicators and not to irregularities in the fuel pump, and in each case the results were the same.

Interpretation of Diagram.—In each of the foregoing examples of diagram error the error is due to mechanical faults in the indicator mechanism, but in nearly every case a superficial diagnosis would ascribe the error to malfunctioning of the engine. Interpretation of diagrams in terms of engine performance has received considerable attention, but the man in the engine room is likely to overlook the limitations of the indicator and its operating mechanism and assume that the diagram is a true picture of what is occurring within the cylinder. With a proper understanding of the probability of mechanical errors in the indicator mechanism, a more rational interpretation of the indicator diagram is possible.

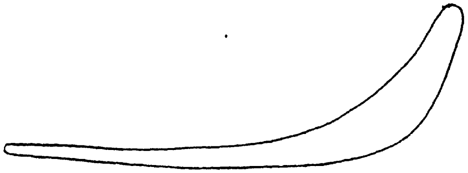


FIG. 372.—Normal indicator diagram.

The engineer should appreciate the possibilities of the indicator as a means of locating faults in the engine, but at the same time he should understand that interpretation of the diagram may not be correct. It is a common assumption that if the flow of fuel through the spray valve of an air-injection Diesel is insufficiently restricted, it will be blown into the cylinder too suddenly, causing too rapid combustion and considerable rise in pressure above compression pressure, and the indicator diagram would be expected to show this sudden pressure rise. That this is not always true is shown by Fig. 372, which shows a normal indicator diagram and Fig. 373 taken from the same cylinder under the same conditions except that all the atomizer disks were removed from the spray valve. In the latter case the diagram obtained might be interpreted to show very slow admission of fuel, due to too much restriction in the oil flow path or too low spray air pressure. As a matter of fact the fuel is blown into the cylinder in an unsufficiently atomized condition, so that it cannot mix thoroughly with the air. As a result combustion is too slow, the combustion line droops, and the fuel is not completely burned until late in the stroke.

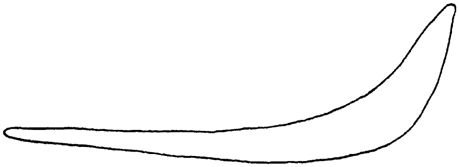


FIG. 373.—Result of removing atomizer discs.

Figures 374 and 375 show very clearly the effect produced in the indicator diagram by varying the spray air pressure. Cards were taken at every 50-lb. variation in pressure, but the two shown represent the extreme limits, 374 being taken when the air pressure was 850 lb. and 375 when it was 700 lb. It should be noted that the latter diagram presents about the same appearance as would be produced by a very much delayed opening of the spray valve. This indicates the possibility of a wrong diagnosis when a card of this sort is taken.

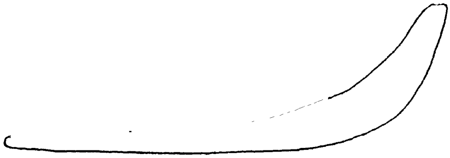


FIG. 374.—Diagram with normal air pressure.

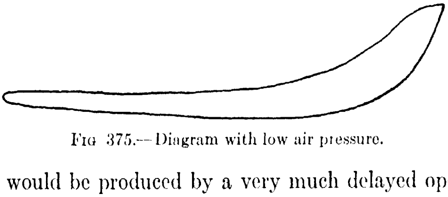


FIG. 375.—Diagram with low air pressure.

It would be impossible to cover all the various changes that will occur in character of diagram with changing conditions in the indicator or in the engine. Variations in fuel viscosity and temperature; valve timing, air pressure, spray-valve adjustment, engine speed, compression, condition of the indicator, etc., all affect the shape of the indicator diagram, but the few examples cited, which are reproductions of actual cards taken under operating conditions, indicate the sort of study that is required to be able to recognize the difference between the effects of these various influences.

Study of the Diagram.—The expansion and compression lines on the indicator diagram will not follow the adiabatic lines for air. This is due to the fact that the gases in the cylinder during expansion are not pure air but a mixture having a variable specific heat. In addition, there is a loss of heat to the cylinder walls during the early part of the stroke and an absorption of heat from the walls during the latter part. These combine to cause the expansion curve to follow the law PV^n = constant where n ranges from 1.25 to 1.40. On the compression stroke the air being compressed loses heat to the walls, causing the exponent n to be around 1.35 but in certain engines it may even exceed 1.41.

Regardless of what the exponent n for the expansion may be, it should be the same during the entire expansion period. If n has a lower value, say 1.25, during the early part of the expansion followed by a steeper portion of the curve with n equal to 1.3 to 1.35, it will be found, in most instances, that combustion was not completed during the period of fuel injection and *afterburning* occurred during the expansion.

Use of Logarithmic Chart.—It is quite impossible to detect a change in the curve as shown on the indicator diagram. However, if the indicator diagram be transferred to a logarithmic paper, afterburning can be detected. This is due to the fact that on logarithmic paper a curve PV^n = constant becomes a straight line, and if n varies then the line is no longer straight but is curved.

To make the transformation, logarithmic paper, Fig. 376, is purchased at an architect's supply house. If this is not possible, the logarithmic paper may be drawn by using the logarithmic scale on a slide rule a shown at Fig. 377. The scale is laid horizontally and division made on the paper beginning at the 0.8 point as outlined. This point can be called 0.08 or 0.8 as the engine cylinder volume may dictate, followed by 0.09 (or 0.9), 0.1 (or 1), 0.15 (or 1.5), etc. The vertical scale is laid out in a similar

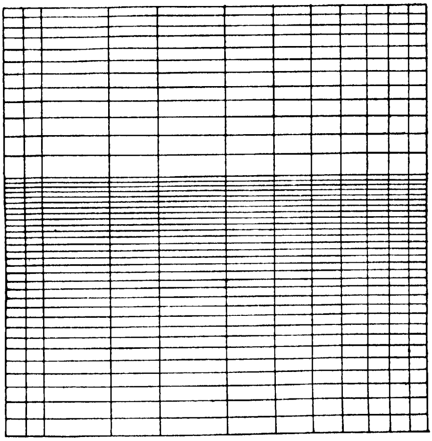


FIG. 376.—Logarthmic paper.

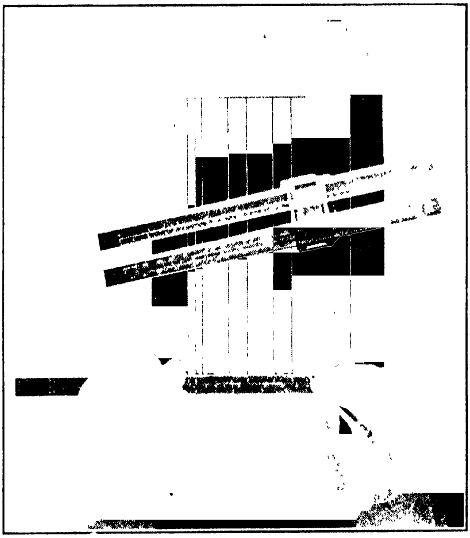


FIG. 377.—Making logarthmic paper by using slide rule.

manner, beginning at the zero point, and calling the unit or 1 point 100 lb., the 2 point 200 lb., etc. Crosslines are drawn, resulting in a cross-section chart, as shown in Fig. 376. Purchase paper has the divisions made by starting at 1, then 2, 3, etc., instead of 0.8.

The indicator diagram is now divided into volume and pressure units. It is not necessary to divide the volume dimension into units representing the actual cylinder volume but the division may be into units of tenths, or of parts of one hundred, making the total diagram volume plus the clearance volume equal to unity or one hundred. The system of division is shown in Fig. 378.

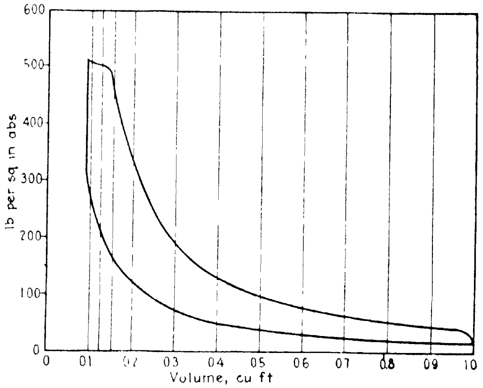


FIG. 378.—Dividing indicator diagram.

It is necessary to have the clearance volume exact or the logarithmic diagram will be worthless. In a vertical engine with a smooth clearance volume the linear dimension will represent the volume close enough. In an engine with a cylinder head similar to Fig. 330 it will be necessary to measure the volume. This may be done by putting the engine on dead-center, placing wax about the piston top to prevent leakage along the piston. The combustion chamber is then filled with water out of a weighed bucket, until the chamber is completely filled. The amount of water used is found by reweighing the bucket and finding the difference in weights.

Upon transferring the diagrams, if the clearance volume has been accurately determined, the compression line should be straight, as in Fig. 379. If the expansion line shows an angle as in line *abc*, Fig. 379, it may be assumed that there is some afterburning. In this chart the expansion lines *ec*, *fc*, and *gc* are fairly straight, incidentally the diagrams shown were in some instances taken from solid-injection engines.

Engineers who find that the exhaust temperatures are running high or that the exhaust valves burn, will upon such transformations to the logarithmic paper discover that the expansion lines show afterburning. Such indicator work may appear laborious,

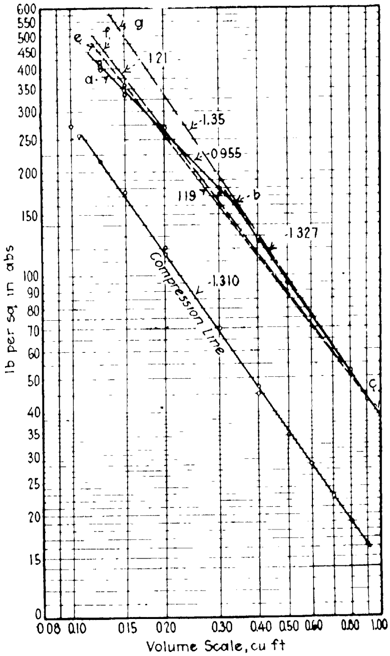


Fig. 379.—Compression and expansion lines on logarthmic paper.

but every engineer should use an indicator and he will find that the study of the logarithmic diagram of value.

Indicating the Solid-injection Engine.—The method of indicating this engine is quite the same as with the Diesel to discussed in this chapter. The indicator diagram for the solid-or pump-injection engine, however, does not show the constant pressure combustion line. Combustion is at both constant volume and constant or falling pressure. The appearance of the diagram is, then, quite different, as will be seen by a study of the diagrams of the several engines.

A diagram of a P-R engine is shown in Fig. 380. The peak *a* shows that the explosion occurred after dead-center.

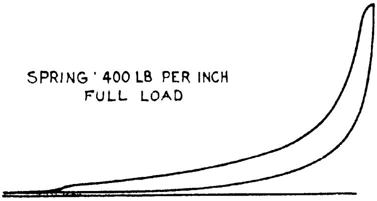


FIG. 380.—Indicator diagram of solid injection engine.

Before going into the method of finding the horsepower developed within the engine cylinder by measuring the indicator diagram, it is well to review the meaning of *force, work* and *power.*

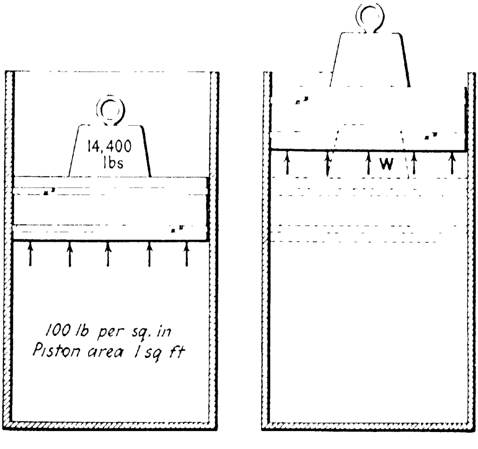


FIG. 381.—Work being done in lifting weight.

Force.—*Force* may well be defined as that which either causes motion or tends to either cause or prevent motion. Suppose we have a cylinder, Fig. 381, filled with a gas at some pressure such as 100 lb. per square inch and the cylinder fitted with a frictionless piston. The gas, if not opposed by some external force, would expand and push the piston out of the cylinder. If the cylinder had a cross-sectional area of 1 sq. ft. (144 sq. in.), then to prevent the expansion of this gas we must load the piston with 144 × 100 or 14,400 lb. This weight just equals the pressure exerted against the piston by the gas. The gas pressure is evidently, by definition, a force since it tends to push the piston, and likewise the attraction of gravity on the weight is also a force since it tends to push the piston inwardly. These two forms are opposite and equal, and as long as this condition exists the piston will not move.

Work.—If the amount of weight holding the piston down be reduced, the force exerted by the gas upon the piston will be greater than the opposing force due to the weight, and the piston will move upward. The force P exerted by the gas over a distance S has then performed work. Work is, then, *force times distance* or may be expressed as $W = FS$. This is similar to what takes place within the engine cylinder, when the pressure-exerting force upon the piston causes the latter to move and by a suitable connecting rod, crank and pulley wheel does work by driving machinery, etc.

Work is equal to force times distance, and, if pounds and feet are the units used, it is measured by a unit called a foot-pound. It will be noticed that the rate at which the force moves does not enter into the amount of work done. There is just as much work done in lifting 100 lb. from the floor to a height of 100 ft. at the rate of 1 ft. per day as there is in lifting it at the rate of 1 ft. per second.

Power.—The element of time enters when considering the *power* developed in doing work. Power is, then, the rate of doing work. James Watt, when installing his steam engine to replace horses in the English mines, wanted some scale of comparison of the engine power with that of the draft horses. In order to have an excess of power he rated his engines on the basis of lifting 550 lb. 60 ft. in one minute or 33,000 lb. a vertical distance of 1 ft. in one minute, this being somewhat more than the mine horses could do. He called this rate of doing work 1 *horsepower.*

Horsepower.—The horsepower is, then, a rate of doing work equal to the lifting of 33,000 lb. 1 ft. per minute. This is likewise equal to a rate of 550 ft.-lb. per second or 1,980,000 ft.-lb. per hour. This is the standard unit employed in English computations.

When the metric system is used, obviously the unit of power is different.

Since the pressures during the suction and discharge strokes are about atmospheric the net pressure for these two strokes is zero. The net average pressure exerted on the piston is obviously the difference between the average pressure on the power stroke and the average pressure of the compression stroke. If this net average pressure be known, the work done per cycle is the product of this pressure per square inch exerted against the piston times the area of the piston in inches (to obtain the total pressure) times the distance the piston is moved by that pressure. The work done per cycle in foot-pounds is then Pla, where

P = m.e.p.,
l = stroke in feet;
a = piston area in square inches.

The power developed in one cylinder per minute is the product of the work Pla and the number of times the piston is forced forward on the power stroke or

$$Power = work\ per\ minute = Plan$$

where n is the number of power strokes of the piston; this for a two-stroke-cycle engine is equal to the number of revolutions, and for a four-stroke-cycle is one-half the revolutions per minute.

Finding Mean Effective Pressure.—The indicator diagram is a closed figure and its height at any point is proportional to the cylinder pressure at that point in the stroke less the cylinder pressure at the same point in the compression stroke. The length is proportional to the engine stroke, and if the area is known the average height will be found; this height is proportional to the mean effective pressure.

To find the mean effective pressure the simplest way to obtain the average height of a closed figure of this kind is to divide its area by its length. If a plane figure is 4 in. long and covers 8 sq. in., it is easy to see that, whatever its shape may be, its average height is $8 \div 4 = 2$ in. This is the way that most indicator diagrams are treated. The diagram area is measured with a planimeter, and by dividing the area by the length the average height is obtained. This height multiplied by the scale of the spring with which the diagram was taken, gives the mean effective pressure.

In the absence of a planimeter the mean effective pressure may be measured approximately by dividing the diagram into equal divisions, as in Fig. 382, measuring the height l of each division and multiplying their average height by the scale of the spring. The more numerous the divisions the more accurate the result. Ten are generally used because it is easy to divide by 10 in getting the average.

It is almost impossible to divide a diagram length directly into 10 even sections since the diagram is seldom of an even measure, as 3 or 4 in. To find the heights l of the 10 divisions, which are the measurements actually desired, as shown in Fig. 382, rather than the divisions themselves, vertical lines perpendicular to the vacuum line at O and X and touching the ends of the diagram to be measured, should be erected. Next, taking a 5-in. rule, place it at an angle with OX that allows the zero reading on the scale to be at O and the 5 mark to cut the vertical line drawn through X the end of the diagram, as shown in Fig. 383. Now draw a line along the rule from O, and using the rule and a pin, start at the ¼-in. scale reading and make a pin mark; next make one opposite the ¾-in. mark, followed by one at 2¼, at 2¾, 3¼, etc., each ½ in. apart. By drawing through these pin marks lines perpendicular to the vacuum line OX the diagram length is divided into 11 spaces, the first one OK' and the last one SX being just one-half the length of each of the remaining

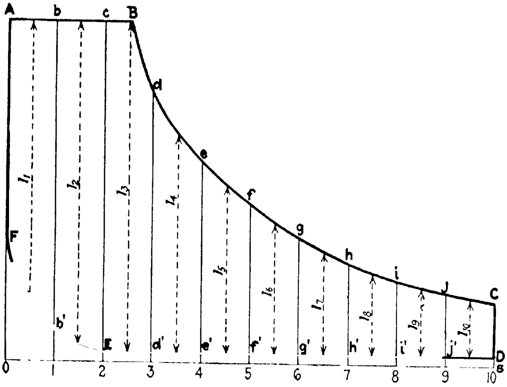


FIG. 382.—Finding mean height of a diagram. (One shown is a steam engine diagram.)

spaces. The points at which these vertical lines cut the upper and lower lines of the diagram define the limits of the several lengths l_1, l_2, l_3 . . . l_{10}.

To find the average height, the sum of l_1, l_2, l_3 . . . l_{10} is first found, and by dividing by the number of lines (10), the average height is determined. This is proportional to the *mean effective pressure*, the factor being the spring scale. If a 200-lb. spring has been used in the indicator, it is best to use a rule with a 40-lb. scale marked on it for measuring the distances l_1, etc. By then dividing by 50, the mean effective pressure in pounds per square inch is found. If such a rule is not at hand an ordinary rule may be used and the value obtained in inches multiplied by the spring scale.

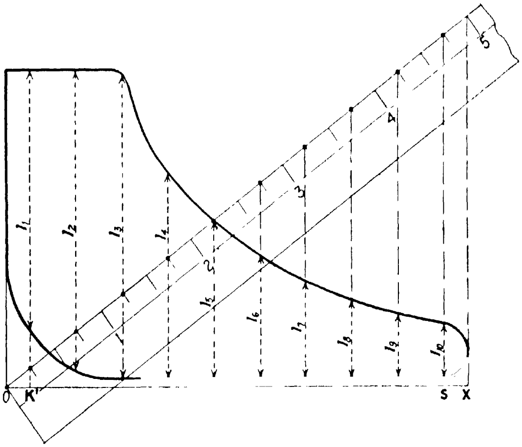


FIG. 383.—Dividing diagram length into sections to find mean height.

A plan followed by many engineers is to take a strip of paper and by laying it along l_1 first mark this length on the paper, then the strip is transferred to l_2 and this added to the length l_1 already marked on the paper strip. By thus adding all the ten lengths, the sum may be measured by a scale, and the error should be less than when the scale is set against each length.

The method here outlined may be placed in the form of a rule as follows:

Divide the diagram length into 11 *divisions, the first and last being one-half each of the other divisions. Erect vertical lines at each division. Add together the distances intercepted on these*

In the absence of a planimeter the mean effective pressure may be measured approximately by dividing the diagram into equal divisions, as in Fig. 382, measuring the height l of each division and multiplying their average height by the scale of the spring. The more numerous the divisions the more accurate the result. Ten are generally used because it is easy to divide by 10 in getting the average.

It is almost impossible to divide a diagram length directly into 10 even sections since the diagram is seldom of an even measure, as 3 or 4 in. To find the heights l of the 10 divisions, which are the measurements actually desired, as shown in Fig. 382, rather than the divisions themselves, vertical lines perpendicular to the vacuum line at O and X and touching the ends of the diagram to be measured, should be erected. Next, taking a 5-in. rule, place it at an angle with OX that allows the zero reading on the scale to be at O and the 5 mark to cut the vertical line drawn through X the end of the diagram, as shown in Fig. 383. Now draw a line along the rule from O, and using the rule and a pin, start at the ¼-in. scale reading and make a pin mark; next make one opposite the ¾-in. mark, followed by one at 2¼, at 2¾, 3¼, etc., each ½ in. apart. By drawing through these pin marks lines perpendicular to the vacuum line OX the diagram length is divided into 11 spaces, the first one OK' and the last one SX being just one-half the length of each of the remaining

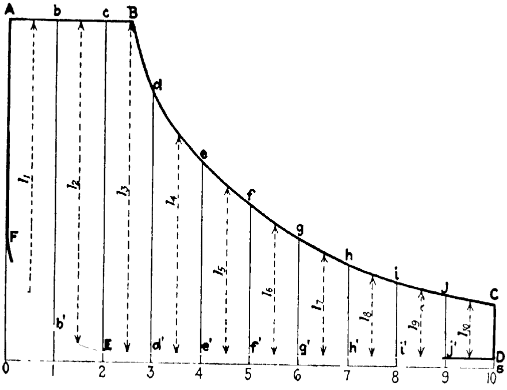


FIG. 382.—Finding mean height of a diagram. (One shown is a steam engine diagram.)

Horsepower at High Altitudes.—Figure 384 is a chart issued by the Nordberg Mfg. Co. which shows the decrease in engine output at localities above sea level. This drop is due to the fact that air at a high altitude weighs less per cubic foot than at sea level. The actual weight of air taken into the cylinder is then less and the amount of fuel burned and engine output is decreased.

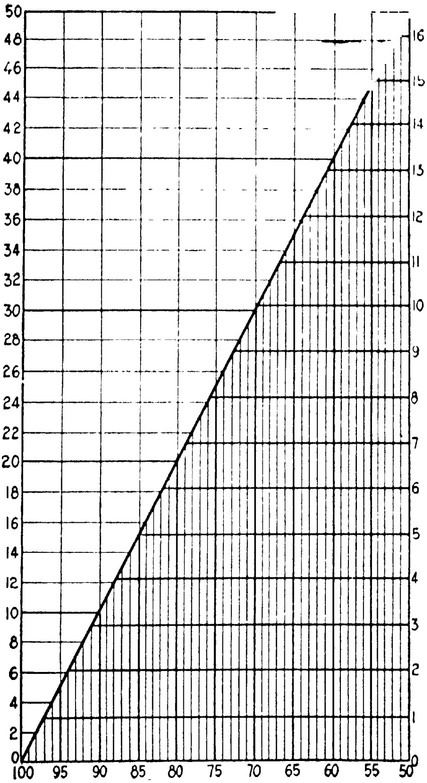


FIG. 384.—Decrease in power with increase in altitude.

Calculation of Probable Mean Effective Pressure.—Robertson Mathews published in *Power* a chart, Fig. 385, by which, assuming an air to oil ratio and a fuel consumption per horsepower, the mean effective pressure may be obtained. The dotted line on the chart will show at once how the chart may be used.

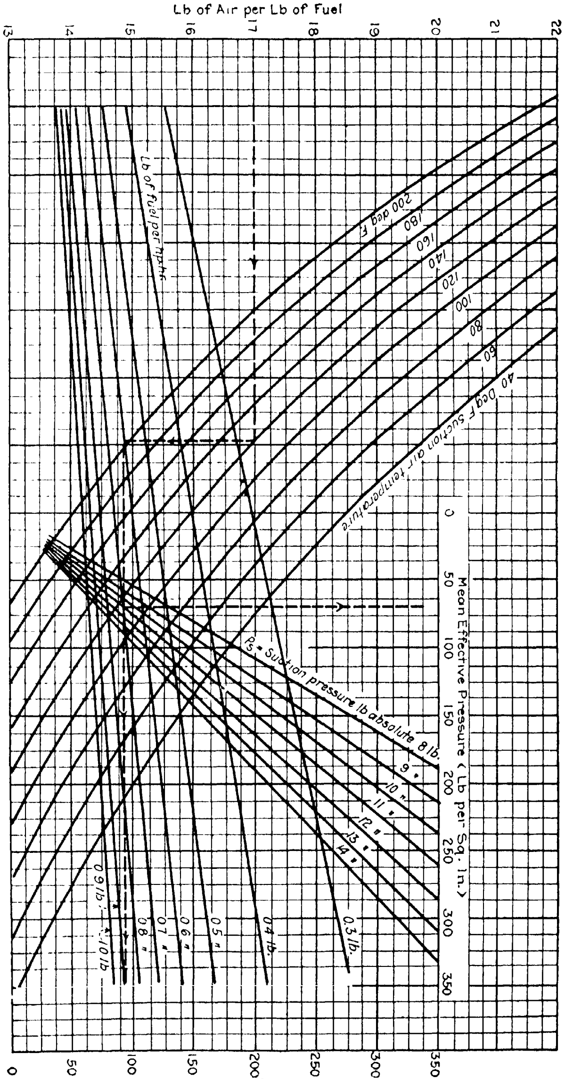


FIG. 385.—Chart for finding m.e.p. when the fuel consumption, air-fuel ratio, and suction pressure are known.

CHAPTER XXII

OPERATING PROCEDURE

Starting.—The method for starting adopted on all Diesels embodies the use of high-pressure air, which by entering one or more cylinders forces the piston downward. The air is discharged at the end of the stroke and at the proper time (that is, the point of fuel injection) another air charge is introduced through a mechanically operated air-starting valve. This procedure is continued until the other cylinder starts firing, whereupon the air supply is shut off.

Starting Procedure.—The exact method of starting differs with the several makes of engines. The general plan is as follows:

1. The engineer should be on duty 15 minutes before starting time to make all necessary preparations.

2. Open and adjust all lubricators and see that main bearings and gears have sufficient oil.

3. Open valve on fuel-oil line, turn the fuel-pump lever on *filling*, after ascertaining that test cocks on cylinder heads are open. As soon as the fuel runs a steady stream *without air bubbles* put the fuel-pump lever in *running position.*

4. Bar engine over one complete turn immediately before starting and then set one of the starting cranks about 15 deg. past top center, after which barring gear should be thrown out of gear and locked.

5. After ascertaining that the pumps are working satisfactorily, the testing cocks should then be closed and 5 to 10 strokes pumped into the spray valves in case the pumps have hand plungers. If fuel pumps have been dismantled, an easy way to ascertain if pump is all right is to put blast air in line and by pumping with hand pump, resistance should be felt which will vary according to the air pressure in the blast line. If no resistance is felt, this would show that the pump is not in working order and it would be useless to attempt starting the engine as fuel pump must be capable of pumping oil against a pressure as high as 1,100 lb. when engine is in service.

6. Turn on the cooling water and never start before same runs freely.

7. Open air-compressor throttle and see that drains on the coolers are closed.

8. Ascertain air pressure in the air bottles. Never start with less than 700 lb. pressure, more preferable, providing it is not more than 1,000 lb. per square inch. Open compressor and blast valves on a blast bottle, then open the main valve on one of the starting bottles, which will set the engine in motion.

9. If the starting and injection pressure is low prevent a false start by taking the following precautions:

Prime the fuel delivery pipes (see specific instructions regarding the fuel pump).

With a bar under the valve levers try all the valves to see that none of them stick.

Fill the water jackets with warm water.

Remove all possible load from the engine.

If the injection-air pressure is also insufficient (below 525 lb. per square inch) the valve of the injection-air bottle should not be opened. The air pump will furnish the required injection-air pressure after a few revolutions. When the engine comes up to speed open the valve in the injection-air bottle gradually to raise the pressure up to normal.

If the engine drives a generator this can be used as a motor for starting, in the following manner:

Hook down the exhaust valve levers.

Do not open the injection-air receiver.

Place the controlling wheel of the fuel pump at "Stop."

When the engine comes up to speed release the exhaust-valve levers and set the fuel pump control wheel to "Work"; prime with kerosene if used for fuel.

Never use oxygen for starting.

10. After the engine has made four or five revolutions, put starting levers on each starting cylinder to the running position which puts starting cams out and fuel cams in action, then close the valve on the starting bottle.

11. As soon as engine is under full speed and the cylinders fire regularly, load may be put on.

Causes of Failure to Ignite.—Failure of the engine to start firing may be traced to one or more of the following troubles, insufficient or no fuel; low compression pressure in the engine cylinder; too low an injection air pressure; cold cylinders.

If the engine fails to fire at the first trial the engineer should not waste the compressed air in repeated trials, but should investigate and make the necessary correction.

Insufficient Fuel.—Insufficient fuel or total failure of fuel supply to cylinder may be caused by an empty fuel service tank or by a closed stop valve in the fuel line between measuring pump and tank, and these should be checked up first. On some types of fuel-measuring pumps the air-starting gear and pump mechanism are not interlocked in such a way that the pumps are automatically put into operation when the engine begins to turn with air. In such a case it may happen that the pump levers are not set in the operating position before starting.

Because of the small quantity of oil handled per stroke by the fuel-measuring pump and the high pressure pumped against, this pump is very sensitive to air that may be present in the oil. A fundamental requirement in good pump design is that no pockets may be permitted in the oil passages in pump or valve chamber, where air might collect, but many pumps have been and are still being built that do contain such pockets. Most pumps are provided with vent valves so that the collected air may be blown out. Sometimes the fuel may be prevented from reaching the pump by an air pocket in the pipe between the pump and the tank.

The most common cause of failure of fuel supply is leakage of air past the check valves in the fuel lines to the spray valves. In all closed-nozzle-type spray valves the mixing chamber in the valve body, where the spray air and oil mix before entering the cylinder, is always in direct communication with the spray-air system and consequently is filled with air at injection pressure. In order for the fuel pump to force the oil into this chamber against the air pressure, it is essential that the oil pipe be full of oil right up to the inlet to the valve chamber, so that when the pump forces a small amount of oil into the pump end of the pipe, an equal amount will be forced out of the other end into the valve chamber.

It is obvious that if this pipe is partly filled with air, the oil column, when acted upon by the charge of oil being forced into the pipe by the pump, will simply compress the air and no oil will be discharged into the valve. If the discharge valve of the fuel pump is perfectly tight, no air from the spray-valve chamber can force its way into the oil pipe after the pipe is completely filled with oil, but the fine grit present in nearly all fuel oil makes it very difficult to keep this valve perfectly tight very long. For this reason practically all Diesel-engine builders install a check valve in the oil line to each spray valve, as close as possible to the point of entry of oil into the spray-valve body. This valve closes against the air pressure in the spray-valve body so that the oil column in the pipe is subjected to pressure only during the time the pump is discharging into the line at one end and forcing the oil through the check valve at the other. With this arrangement a pump will work quite satisfactorily even though the discharge valve is not perfectly tight, as long as the check valve remains tight.

Testing the Spray Valve.—This valve can be tested while the engine is stopped, by turning spray air from the bottles into the air line to the spray valve and then opening the by-pass in the fuel-oil line near the check valve. If the valve leaks, the air will blow out of the by-pass. If there is no by-pass in the line, the oil pipe may be disconnected at the pump and the air will blow out there. Before making this test, the engine must be jacked around until the spray valve, to which is attached the line being tested, is in the closed position so that the spray air will not blow into the cylinder. In order to provide additional insurance against spray-air leakage into the fuel-oil lines, some builders provide two check valves in each oil line and two discharge valves in each pump.

Leaky suction valves in the fuel pumps, or valves stuck open, may be responsible for the failure of the oil to reach the cylinders. Examination of the valve and seats will usually indicate a leaky condition.

Compression Too Low.—If the compression in the cylinders is not high enough, the temperature of the air in the cylinders will be too low to ignite the oil, as shown in Fig. 372. If the compression is due to leaky cylinder head or valve-cage gaskets, or leaky relief valves, the leakage will be indicated by the noise of the escaping air. Other causes are leaky inlet, exhaust or air-starting valves, and stuck piston rings. The methods of remedying these defects are obvious and will not be further enlarged upon.

It is seldom that wear of the piston or of the rings takes place so suddenly as to cause ignition failure without due notice. With worn pistons the engine will become harder to start until it finally refuses altogether.

Fuel Injection Too Late.—If the fuel is not injected into the cylinders until after the compressed air has started re-expanding as the pistons move away from the heads and increase the cylinder volume, the temperature of the air may have fallen so low that it will not ignite the oil, and the effect produced is the same as in the case of low compression. This late injection may be caused by the adjustable nose on the spray-valve operating cams slipping. The clearance between cams and rollers may be too great or the valves may be clogged so that the fuel does not flow rapidly enough. The cams should be examined to see if they have slipped on the shaft; if they have not, then the cam toes need advancing by means of the adjusting screws. The rollers should be examined to see if any are badly worn or broken. Each valve should be checked with the dial plate or the valve-setting marks on the flywheel.

If the spray valves are clogged so that the fuel is retarded in its passage through the valves, an abnormal rise in spray-air pressure will be noted if the compressor suction is open wide when the engine is turning on starting air. The valves should be disassembled and all sediment and carbonized oil that may have accumulated cleaned out.

Failure of Injection Air.—The capacity of the injection-air bottles is often so small that if the injection-air compressor does not begin charging immediately upon starting the engine, the result will either be complete ignition failure or ignition will occur for a few revolutions, then fail as the pressure in the bottles falls. When this occurs, no further attempts to start should be made until the compressor trouble is located and remedied.

The most common causes of loss of compressor capacity are broken or leaky valves. The location of the defective valve may be determined by observing the gage pressures in the different stages while the engine is turning over. An abnormal rise in pressure in the first or second stage indicates that air is leaking back through the discharge valve in that stage. Rise of pressure in the high stage may indicate a closed stop valve in the discharge line to the engine, clogged strainers or clogged spray valves.

If an excessive amount of lubricating oil is used in the compressor, a jelly-like emulsion will be formed, which will lodge in the strainers and interfere with air flow. If the compressor shows loss of capacity, with pressure below normal in all stages, it may be due to obstruction of the suction of the first stage. In the case of compressors that are regulated by throttling, this suction loss of capacity may be found to be due to the suction valve being closed. Another cause for rapid loss of spray-air pressure is sticking of spray valves. If a spray-valve stem jams in its guide so that the valve is not forced back to its seat by its spring, the spray air will blow into the cylinder during the whole cycle and so much air will be blown away that the pressure in the system will fail. A condition of this kind will make itself known by very severe explosions in the affected cylinder, due to preignition of fuel that has been blown into the cylinder too early in the cycle.

Cylinders Too Cold.—If the jacket-water circulating pump is started before the engine, it may happen that the cylinder walls and cylinder heads may be chilled to the point where ignition is interfered with, this condition being most likely in cold climates, during the winter months when the cooling-water temperature is very low. This cooling affects the ignition in two ways it reduces the temperature of the compressed air in the engine cylinders and it also increases the viscosity of the fuel oil after it is deposited in the spray-valve cavity, so that atomization of the oil is more difficult and its passage through the valves is retarded. In cases where this trouble is experienced, it is best not to start the cooling-water circulating pump until after the engine is started. When steam is available, it is advisable to make a connection to the water system so that the circulating water may be heated and the cylinders warmed up before starting the engine.

Loss of Compression.—The most frequent causes of loss of compression are leakage of air out of the cylinders, due to leaky exhaust valves, inlet valves, relief valves and air-starting valves; leaky cylinder-head gaskets; and broken, stuck or worn piston rings. In the case of two-stroke-cycle engines of the valve-scavenging type, leaky scavenging valves may be added to this list. When the trouble is caused by leaky relief valves or leaky cylinder-head gaskets, no special investigation is required to locate the leaks, as they will make themselves known by the noise of the escaping air. A leaky air-starting valve can be located by feeling the air pipe below the valve; if the valve leaks, the pipe will be hot. If the inlet valves draw the air in through individual pipes, a leaky valve will make its condition known by a loud whistling noise in the pipe attached to that particular valve, but where all the inlet valves take in air through a common pipe or manifold, the indentification of the leaky one is not so easy. Leaky exhaust valves are revealed by a whistle in the exhaust line. In that form of construction characteristic of most large, crosshead-type engines, where the lower ends of the power cylinders are not inclosed within the housing, piston-ring leakage makes itself known immediately by the peculiar noises, varying in intensity from a sneeze to a loud explosion, made by the air and the products of combustion blowing past the piston rings and out of the open ends of the cylinders. In the case of trunk-piston engines, where the lower ends of the cylinders are inclosed within the housing, the noise can seldom be heard and the piston rings may leak for a long time without the knowledge of the engineer, unless indicator diagrams are taken or the rings examined, although leakage past the piston will always reveal itself by the smell of the crankcase when opened. If leakage past the piston rings is occurring, the normally bright wearing surface of each ring will be blackened at the points of leakage, and the blast effect of the gases will sometimes create a stratified appearance on the surface of the piston.

When low compression in one or more cylinders is shown by the indicator diagrams, circumstances will usually indicate the nature of the cause. If the engine has just been erected and there is no reason to suspect leakage, it is safe to assume that the linear clearance between pistons and cylinder heads has not been correctly adjusted; on the other hand, if the engine has been in service any length of time, the trouble will be found to be due to leakage at one or more of the points enumerated, unless the bearings have been adjusted frequently, in which case the decrease in compression will occur gradually. If leakage is causing the trouble, the point of leakage may be determined by a process of elimination. In most cases it will be found that the trouble is caused by leaky exhaust valves or piston rings. If any valve or cylinder-head gasket is found to be leaking, the procedure for correcting the trouble is obvious and need not be explained.

Leaks past the piston rings that are caused by wear of rings or liners may sometimes be stopped by peening the inside surfaces of the rings, to increase their tension. This peening should be done with a ball-peen hammer while the ring is held on a smooth metal surface, striking light blows with the hammer at a point directly above the point of contact of the ring and the plane surface, and rolling the ring until its entire inner surface has been peened.

If the cylinder liners are worn oval, the only remedy is to rebore the liners or install new ones. The rules of the classification societies require that a number of spare piston rings be carried on all Diesel-motored ships, so that new rings are always available for replacing broken ones, and all well-equipped stationary plants carry an extra set. If low compression is due to excessive clearance between a piston and its cylinder head, this clearance must be decreased by lengthening the piston, piston rod, crosshead and connecting-rod assembly.

The only practicable method for doing this is to add sheet-metal shims between the foot of the connecting rod and the top shell of the crankpin bearing. The engine builder usually furnishes a table showing the thickness of shims required for changing the compression any desired amount within the range of pressures that may be expected in practice. Having determined from the indicator card the amount it is desired to increase the compression and prepared shims of the necessary thickness, the crank to be worked on is placed near top center, and the weight of the connecting rod, crosshead, piston rod and piston supported by a bar placed through the guide and under the crosshead slipper, as in Fig. 133, or by some other suitable means.

The crankpin nuts are then removed, the shaft jacked around until the top box is pulled away from the connecting rod, the lower box and crankpin bolts lowered away until the bolts are pulled out of the holes in the connecting rod, and the shims may then be slipped into place. This job may be greatly simplified if the shims are cut away on a radius. If this is done, the crankpin nuts need not be entirely removed from their bolts and no support for the lower box need be rigged; the nuts are simply slacked off, the box pulled away from the rod enough to admit the shims and when the centering lug has been slipped through the center hole in the shims, the latter can be revolved into place.

If the engine is of the trunk-piston type, the piston can be supported as shown in Fig. 133. Two timbers, a trifle shorter than the distance between the piston and the blocking alongside the main bearings, are inserted when the crank is on top dead-center. Turning the flywheel will allow the weight of the piston and rod to rest on the timbers. After loosening the rod bolts, the wheel may be turned until the crankpin bearing leaves the big end sufficient for the insertion of the shims.

To Stop Engine.—Before stopping engine be sure that all air bottles are charged up to full pressure.

If engine has been working on heavy fuel, light fuel must be turned on from five to ten minutes before shutting down.

To stop engine set the control of the fuel pumps to "Stop" by means of lever provided for this purpose, which puts fuel pump out of action. Next, close the valve on the blast bottle and, when the engine has come to a stop, close the compressor valve and put starting levers in to "Start." Release compression by means of levers acting on valve rockers, or by opening indicator cocks on smaller engines. Also open drains on compressor coolers. The cooling water should be allowed to run for 10 to 30 minutes. This will prevent the seizing of the piston or pin.

Operation.—Great care should be exercised with a view of keeping the exhaust clean. The usual causes for smoke are as follows:

Engine being overloaded.

Fuel-lever rollers having none or too much clearance.

Blast pressure being too low.

Presence of dirt in atomizers.

Scavenger valves or fuel valves not closing properly.

Use of unsuitable fuel.

Load on cylinders is not equally divided.

Compression is too low. (This should be between 450 and 500 lb.)

The Engine Knocks or Pounds in Operation.—Heavy knocks and pounding are caused by slackness in the crankpin box or in the piston-pin box or by explosions in the cylinder due to a fault in the ordinary process of combustion.

Bearing knocks are located as follows:

Stop the engine, turn it to the upper dead-center and place a lever under the crankpin box. With a finger partly on the box and partly on the shaft any slight play in the bearing may be noted.

Looseness in the piston-pin box is detected by holding the connecting rod firmly, raising the piston and observing the vertical movement between the piston and cylinder. This is more easily observed when the piston is at the lower dead center.

Explosion shocks are due, as already mentioned, to a fault in the process of combustion; it may be traced to one of the following causes:

(*a*) The fuel valve opens too soon, indicated by a sharp noise in the cylinder. The fuel valve opens too late, indicated by a dull thud in the cylinder.

(*b*) The injection-air pressure is too high.

(*c*) The needle does not seat properly, allowing fuel oil to get into the cylinder during the suction and compression strokes.

(*d*) Fuel oil enters the cylinder in large quantities, because the cone of the pulverizer is not in good condition.

Cooling Water.—The outlet temperature of the cooling water should be kept between 130 and 140 deg. F. on cylinders, 110 to 120 deg. F. on pistons and 140 to 150 deg. F. on exhaust, but under no condition should the latter figure be exceeded. In cases where hard water is used for cooling, the above temperature should be decreased.

The Engine Smokes When Running.—This may be due to:

Too heavy load.

Too low injection pressure.

Fuel-valve needle opens too late.

Leaky valves, which reduce the compression pressure.

Fuel-valve burner plate is clogged.

The piston rings stick.

The Speed of the Engine Falls below Normal.—This happens when the engine is overloaded, also in case there is poor combustion. If a bearing runs hot the speed will drop.

The governor does not act properly, for example the regulating collar may fit too tight on the shaft, which lowers the speed.

Air may have forced its way into the fuel pump so that the fuel is not delivered regularly or the governor may be too stiff; so that it does not regulate quickly enough for a variable load.

Air Pressure.—The blast pressure should be varied according to load; about 600 lb. for no load and about 1,000 lb. for full load. If the blast pressure is too low, the engine will smoke, in other words the combustion is imperfect. If too high the engine will knock. Under no circumstances should the blast pressure be under 600 lb.

Condensed moisture from the air should frequently be blown off from the drain cocks on low and intermediate air-compressor receivers, also from the valves provided on the starting and blast bottles.

Care of Valves.—Suction and exhaust valves on large engines should be examined and ground every 2,000 hours, or as soon as same shows signs of leaks. The frequency of inspection of the above valves depends greatly on local conditions.

Fuel-valve needles should be ground and atomizers cleaned and ground every 1,000 hours, or earlier if found necessary.

Air-compressor valves need very little attention but should be examined as soon as compressor shows any signs of losing its efficiency. A good way of ascertaining whether or not valves need attention, is to watch gauges of the different stages as it can easily be seen if there is any leaks from one stage to another.

Fuel-pump valves should be ground once every 1,500 to 2,000 hours and pump washed with coal oil. The necessity of examining these valves depends greatly on the quality of fuel oil which is being used.

Piston.—The main and compressor pistons should be drawn once each year for cleaning or as soon as there are any signs of leaks. Piston rings should be replaced where found necessary and cylinder lubricating system cleaned.

Roller Clearances.—The different valve levers roller clearances should be checked over frequently, as well as every time that levers are removed. The clearances of the rollers may be obtained from the engine builders.

For particulars as to care and adjustment of any engine part, the reader is referred to the chapter dealing with the detail in question.

CHAPTER XXIII

TESTING OF OIL ENGINES

General.—There have been in the past many test codes issued by various engineering bodies, all of these test codes being quite similar. The American Society of Mechanical Engineers has had a committee engaged for some time in drafting a new test code. While the committee has issued only a tentative code which has not been adopted finally by the A. S. M. E., nevertheless there is every reason to believe that the code as reported will be adopted with but few changes. Consequently, the author here reproduces the preliminary draft of this code with a feeling that it is a little more complete than any of the other existing ones.

1. Tests for determining the performance of internal-combustion engines by this code apply to all forms of internal combustion engines, but are limited to the engines alone Separately driven auxiliaries that are essential to the operation of the engine, such as scavenging pumps or blowers and injection air compressors, for example, must be included in the tests, but the testing of such units must follow the other codes appropriate to the unit to be tested and the results in horsepower absorbed introduced and used in this code.

2. Internal-combustion engines, compared to others for which codes are available, include a much greater variety in form, size, speed, weight, fuels and other operating conditions, with more or less special features for particular applications. This fact makes it necessary in a single inclusive code like this to leave many details of the test to the judgment of the test engineer, or to be made the subject of expressed agreement before the test is started. Suggestions are offered in this code as to limitations, alternatives and difficulties to aid in making selections of details, always with the understanding that suitability to the case shall be the basis of judgment.

3. Sizes of internal-combustion engines coming within this code range from 1 hp. to several thousand. Speeds range from less than 100 r.p.m. to over 3,000 r.p.m. Numbers of cylinders range from one to twelve or more; some horizontal and others vertical; some single-acting and others double-acting; some two-cycle and others four-cycle, with every variety of attached and separate scavenging pump for two-cycle engines. Fuels range from the lightest distillates to the heavy residue fuel oils, organic and mineral in the liquid-fuel class, and in the gaseous form include every combustible gas commercially available. Purpose and variety of load or other service conditions normal to and proper for each engine cover equally wide ranges. Stationary engines operate at all loads normally at constant

speed under governor control, for any period of time and any degree of intermittence of use. Marine engines operate normally at full load, and the constant speed fixed by the propeller, but must be maneuvered ahead and astern at any speed under hand control, and may operate for any period of time from short intermittent runs to the continuous full load for a month or more required of seagoing motorships. Aircraft engines must meet about the same service conditions as marine engines, except that a smaller length of single run is required, but they must operate in the greatest known ranges of atmospheric pressure and of temperature Other engines of the automotive group, those for land transportation, including those for automobiles, motor trucks, tractors, railroad locomotives and self-contained cars, are essentially variable in speed as normally used, and equally variable in torque, horsepower being as much a matter of secondary importance as in a hoisting engine or steam locomotive The engines of this last group drive their vehicles through gear sets of different speed ratios, at changing vehicle speeds and changing vehicle resistance, and tests of them must be conducted with due regard to these service conditions.

It is obviously impossible in any single code to prescribe tests in detail that could be equally applicable to all these conditions, so that some exclusions are necessary At the same time it is highly desirable that tests for all internal-combustion engines set by this code should have as much in common as may be possible, even if some latitude of judgment is necessary to make this possible.

4. Excluded from this code are the following

(*a*) Complete internal-combustion engine stationary power plants using gaseous or liquid fuels, which have power-generating or power-absorbing auxiliaries separate from the engine

(*b*) Motor-boat and motorship power plants and the vessels themselves.

(*c*) Aircraft and their power plants.

(*d*) Land propelled vehicles, with or without speed-change gears, including motorcycles, automobiles, motor trucks, tractors and rail locomotives or self-contained cars.

(*e*) All internal-combustion engine "sets," consisting of an internal-combustion engine so combined with a driven unit as to make it impossible to determine separately the performance of the engine itself, or such as falls within the scope of other codes, excepting only electric generating sets. Among these sets herein excluded are internal-combustion-engine pumps, compressors and hoisting outfits, whether used for hoisting or for hauling

(*f*) All internal-combustion engines, the construction or operating conditions of which prevent an output determination in terms of either brake horsepower or kilowatts, for a generating set

5. Included within this code are the following internal-combustion engines not excluded by any of the items of Par. 4.

(*a*) Stationary engines, and electric-generating sets even if not intended for stationary use, burning gaseous or liquid fuels, including the scavenging pumps of two-cycle engines, and injection air compressors, if used and separately driven.

(*b*) Marine engines exclusive of propeller, and burning light or heavy liquid fuels.

(*c*) Aircraft engines limited to block tests, exclusive of propeller, and burning light liquid fuels.

(*d*) Land vehicle engines limited to block tests and burning light and heavy fuels.

6. Reference is made, in connection with the exclusions of Par. 4, to the separate codes for testing:

(*a*) Displacement pumps.
(*b*) Centrifugal and rotary pumps.
(*c*) Reciprocating displacement compressors and blowers
(*d*) Centrifugal and turbo-compressors and blowers.
(*e*) Refrigerating machines and plants.
(*f*) Locomotives.

Reference is also made in general to other codes prescribing items forming part of this code, which must be considered in connection with it. All directions in these codes, noted below, applicable to this one must be followed:

(*g*) General instructions.
(*h*) Definitions and values.
(*i*) Instruments and apparatus.
(*j*) Fuels.

OBJECT

7. In accordance with the "General Instructions," the object of the test, and if more than one, then each separately, should be determined and recorded.

This code is limited to tests in which the object is of a commercial nature. It excludes all tests forming part of experimental investigations, research or development. But while no engineer undertaking tests of a research or development nature needs any code if he is competent to make such investigations, it will usually be found that parts of such research work will be aided and in no way hampered by following parts of this code. It is also in the interests of those branches of engineering, whose progress is based on experimental data, if any parts of an investigation are carried out in accordance with the code, and recorded in accordance with the code. Results so obtained are more easily comparable with the large numbers of tests always being carried out on the lines of the code. The research data then enter readily into the general body of knowledge of the internal-combustion engine.

8. Tests of the kind contemplated by this code and for which it has been prepared, are those in which the object of the test is *definition* of an engine by its performance characteristics.

9. When the object of the test is wholly or in part the determination of fulfillment of a contract guarantee, an agreement should be made before the test between the interested parties on all matters that involve any possible elements of uncertainty, or that may later be a cause of dispute, as noted in Par. 3 of the "General Instructions," and the points agreed upon should be stated in the report of the test.

10. Agreements between interested parties should also be made and recorded even when a contract guarantee is not involved, before undertaking any test in which an element of judgment is involved in the selection of any

instrument, piece of apparatus, physical constant, test method, operation condition or other matter that will promote the accuracy, the value, conclusiveness or the general acceptability of the results when obtained. This matter is one of special importance in tests on internal-combustion engines because of their great variety.

11. Sub-objects of the main object may often be of value in improving the value of results obtained in the test for the main object and may themselves be made the subject of separate tests, but in all cases *only* when made the subject of previous agreement as to the nature of the sub-objects and the test procedure. Sub-objects of a test must be separately reported as such to clearly distinguish them from the main object, and it is advisable that reasons be given for making such special tests. This is especially important when the special test for the sub-object is to be conducted at the same time as the test for the main object. Some sub-objects that may be expected to arise in connection with code tests of internal-combustion engines are listed below, but no special instructions concerning them are included in this code. Selection is left entirely to the judgment and agreement of the test engineers, but none of these sub-objects will be considered as tests made "according to the A. S. M. E code," unless specifically added to the requirements by agreement and so marked in the report. It is recommended as good practice that they conform as far as possible to such parts of this and other codes as may be found applicable

(*a*) Exhaust-gas analysis, with special reference to the determination of air not utilized by the engine in combustion, and of unburned fuel or combustible.

(*b*) Exhaust-gas temperature, with special precaution to get accurate results in view of high velocity of gases, intermittent flow, and cooling conditions in exhaust ports and pipes, determined with special reference to equalization of load between cylinders of multi-cylinder engines, or accuracy of adjustment of combustion conditions, or percentage of fuel heat discharged in the exhaust.

(*c*) Amount of air taken into an engine, with special reference to the volumetric efficiency and the completeness or incompleteness of the charging as a factor in actual as compared to possible power developed.

(*d*) Properties of liquid or gaseous fuels with reference to suitability for use in the engine, or with reference to identity of the fuel as to some specification.

(*e*) Determination of variations in the quality of lubricating oil or its contamination by use, especially by fuel or fuel products, and particularly for engines provided with a fixed amount of lubricant successively reused.

(*f*) Nature of solid deposits in combustion chambers, on pistons, valves and ports, resulting from use and originating in fuel or lubricant or in solids carried by the air.

(*g*) Amount of air required to start an engine provided with air-starting equipment, in relation to the capacity of starting compressors, especially for engines that must be frequently maneuvered in service.

(*h*) Electrical starting requirements of engines using electricity for starting on heaters or electric motors, in relation to generator and storage battery.

(*i*) Amount and pressure of air required for air-spraying apparatus of air-injection oil engines using fuel oils, in relation to the spray-air compressor capacity.

(*j*) Power consumed by each or all of normally attached auxiliaries, with or without other operative characteristics of these auxiliaries, which include electric generators or magnetos, storage batteries or complete electric ignition apparatus, jacket-circulating pumps, direct or radiator cooling fans, lubricating-oil and fuel-oil pumps, starting and spray-air compressors, scavenging pumps of two-cycle engines, attached fuel and lubricating-oil heaters and coolers.

(*k*) Smoke quantity and color or density, and particularly when smokelessness in some degree is a matter of importance or of specification.

MEASUREMENTS

12. The measurements that must be made in a performance test of an internal-combustion engine will be the following quantities:

(*a*) Cylinder diameter, stroke and clearance volume of working cylinders.

(*b*) Diameters of piston rods and tail rods of working cylinders of double-acting engines.

(*c*) The brake-horsepower or shaft-horsepower output.

(*d*) The kilowatt output if engine is direct-connected to a generator.

(*e*) The speed in revolutions per minute

(*f*) The horsepower to drive the water-cooling pump and fuel-pump (if any).

(*g*) The horsepower to drive the independent scavenging pump or blower.

(*h*) The horsepower to drive the independent injection-air compressor.

(*i*) The amount of fuel supplied: cubic feet of gas for gas-burning engines, or pounds of liquid fuel for liquid-fuel engines. If more than one kind at the same time, the amount of each kind.

(*j*) The calorific value of the fuel, high value.

13. The measurements which may be made in addition to those required by Par. 12, will be some or all of the following as may be agreed in advance by the test engineers:

(*a*) Cylinder diameters and strokes of injection air-compressor.

(*b*) Two-cycle engine scavenging-pump diameter and stroke.

(*c*) Diameters of piston and tail rods of scavenging pumps of two-cycle engines.

(*d*) Diameters of piston rods and tail rods of injection-air compressor.

(*e*) The indicated horsepower.

(*f*) The number of explosions per minute, or fuel injections per minute, or in general, combustion times per minute, for each and for all working cylinders.

(*g*) The spray-air pressure of air-injection oil engines.

(*h*) The exhaust back pressure at the engine.

(*i*) The scavenging-air pressure of two-cycle engines, constant or maximum values.

(*j*) The manifold vacuum of carburetor engines.

(*k*) The jacket-water or oil-supply pressure at the main, and if jackets are divided with separate supplies, the pressure at the supply point or branch pipe to each.

(*l*) The jacket-water or oil back pressure at the main outlet, or if jackets are divided with separate discharges, the pressure at the outlet of each.

(*m*) The pressure or vacuum or the hydraulic head, positive or negative of the supply of liquid fuel at the carburetor connection of carburetor engines, and at the injection-pump suction of injection oil engines.

(*n*) The pressure or vacuum of the gas supplied to gas-burning engines at the mixing valve connection.

(*o*) The total pressure, static and velocity of cooling-air supply of air-cooled engines.

(*p*) Barometric pressure.

(*q*) Atmospheric humidity.

(*r*) The pressure of lubricating-oil supply to bearings of engines lubricated by pump forced-feed circulation.

(*s*) The temperature of the atmospheric-air supply.

(*t*) The temperature of the fuel for gas at the mixing valve, or for liquid fuel at the carburetor or injection-pump suction

(*u*) The temperature of the jacket water or oil as supplied to the engine as a whole, or if jackets are divided with separate supplies, the temperature of each separate supply

(*v*) The temperature of the jacket water or oil leaving the engine as a whole, or if jackets are divided with separate outlets, the temperature at each outlet.

(*w*) The temperature of the scavenging air of two-cycle engines.

(*x*) Temperature of injection air.

(*y*) The temperature of the lubricating oil supplied to the engine at as many points as there are different supplies of pump-circulated lubricating oil, or in the crankcase of engines without pumps or having pump deliveries that do not permit of temperature measurement.

(*z*) The temperature of the cooling air supplied to an air-cooled engine.

(*aa*) The amount of lubricant consumed, pounds.

(*ab*) The amount of jacket water or oil supplied, pounds, for engines as a whole, and when jackets are divided for each if both supply and discharge are separate.

(*ac*) Amount, pounds or cubic feet, of cooling air supplied to air-cooled engines.

(*ad*) Amount, pounds, of jacket water evaporated in jackets of hopper-cooled engines.

(*ae*) Amount, pounds, of injection water supplied internally to cylinders of some oil engines, as a liquid or as steam.

(*af*) The compression pressure in cylinders when hot and cold, at normal or at reduced speed at wide-open throttle.

14. In preparing a list of measurements from the items of Pars. 12 and 13, to be made preparatory to the planning of the log, and as a guide to their selection, it must be noted that in the above lists of measurements there are provisions for three groups:

(*a*) Measurements required for the determination of results required for this code, namely, those of Par. 12.

(*b*) Measurements to indicate the maintenance or variation of important operating conditions which might be described as the steady state required for the test prescribed, but which measurements do not themselves enter into the results except as specifying conditions under which the results were obtained. For example, this is the case with all jacket water or oil and all lubricating-oil measurements of pressure, temperature and quantity. These are included in the list of Par. 13.

(*c*) Measurements for one or more of the determinations required by such sub-objects listed in Par. 11, as may be made the subject of an agreement by the interested parties. These are included in the list of Par. 13.

INSTRUMENTS AND APPARATUS

15. Directions regarding the application, use and calibration of instruments and apparatus, do not form part of this code Such information is given in the Code on Instruments and Apparatus, supplemented by that on Definitions and Values, in which will also be found statements on scope and limitations of use, and on accuracy.

The instruments and apparatus required for a performance test of an internal-combustion engine as prescribed by this code will include some or all of the following:

(*a*) Scales with or without special auxiliary apparatus for weighing liquid fuel, with or without tanks to be used in coöperation.

(*b*) Gas meters for measuring gaseous fuel, or gas-metering methods with indirect observation apparatus.

(*c*) Gas calorimeters for determining the heating value of gaseous fuels

(*d*) Baumé oil hydrometer for petroleum liquid fuels, and for indirect determination of calorific power.

(*e*) Pressure gages and mercury or water columns for measuring small pressures and vacua.

(*f*) Thermometers.

(*g*) Barometers.

(*h*) Gas-engine and oil-engine indicators for working cylinders and steam-engine indicators or special low-scale springs for cylinders of two-cycle scavenging pumps and low-pressure cylinders of injection-air compressors.

(*i*) Pressure-indicating or recording instruments for compression pressures or injection pressures in cylinders.

(*j*) Planimeters.

(*k*) Tachometers, revolution counters or other apparatus for measuring speed or number of fuel admissions per minute.

(*l*) Absorption dynamometers of appropriate type for the horsepower, speed and torque of the engine to be tested, especially hydraulic brakes and electric dynamometers.

(*m*) Appropriate electrical instruments and apparatus, if the engine is direct connected to a generator, to provide a suitable electrical load and to measure it.

16. In making a selection of instruments suitable for the measurement of quantities required by the performance test under this code, and constituting its main object per Pars. 7 and 8, it is necessary to exercise considerable latitude of judgment in view of the variety of internal-combustion engines noted in Par. 3, to insure proper relation of selected instrument to the conditions of measurement to be made. Some suggestions are offered below as a partial guide to such selection.

17. Indicators cannot be used at all on some internal-combustion engines and in other cases only within limits. In no case can an indicator be regarded as a precision instrument on an internal-combustion engine to the same degree as on a reciprocating steam engine. Some special cases are noted:

(*a*) When the engine speed exceeds 400 r p m. an indicator must not be used for determining the cylinder mean effective pressure of an internal-combustion engine. This excludes it and the determination of indicated horsepower for all engines of Par. 5(*c*) and 5(*d*), and all engines of Pars 5(*a*) and 5(*b*) operating over 400 r p m

(*b*) When the engine speed exceeds 400 r p m optical indicators may be useful as a means of adjustment of valve or spark timing, but have no other value, and are of no use even for such adjustments in the case of multi-cylinder engines.

(*c*) On all hit-and-miss-governed engines the indicator must not be used for determination of the cylinder mean effective pressure for indicated-horsepower determinations, thus excluding such determinations from tests of hit-and-miss-governed engines

(*d*) When indicator cards taken even at speeds of 400 r.p m or less are not constant in shape and area over long periods of time while a constant load is maintained, the indicator must be rejected as a means of determining cylinder mean effective pressures though it may be used as a means of adjustment of timing, especially as a guide to equalizing the timing for multi-cylinder engines or the two ends of a double-acting cylinder. The test as to constancy of area shall be ±2 per cent, so that when the largest area differs from the smallest area by 4 per cent or more of their arithmetical mean, all shall be rejected At various odd intervals of time, at least ten diagrams shall be taken on one card in addition to those taken at regular intervals, for comparison as to constancy.

(*e*) When indicator cards taken even at speeds of 400 r p.m. or less are apparently constant in area, according to the conditions (*d*), they may still be useless as a means of determining cylinder mean effective pressure and indicated horsepower, when the cylinder pressures are very high, the spring scale large, 500 lb more or less, and the whole measured card area at the same time is very small Under these conditions the precision will be very low and the indicated horsepower much in error Just what set of conditions will produce a result bad enough to reject must be left to the judgment of the test engineer, with the suggestion that the accuracy of planimetering the smaller area be used as a basis of judgment This condition makes it hopeless to undertake determinations of indicated horsepower for many oil engines of the high-pressure type, developing only small mean effective pressures.

(*f*) Even in the cases where the indicator cannot properly be used as an instrument for measuring mean cylinder pressures and indicated horsepower, it can be usefully employed as a means of adjusting and equalizing the cylinder compression, the adjustment of main valves, the point of ignition in electrically ignited engines, of spray-valve timing of Diesel oil engine, or the injection-pump timing of injection oil engines with direct pump injection.

(*g*) When the conditions of proper use of indicators for mean-effective-pressure determinations are all fulfilled, no indicated-horsepower determination shall be made unless a separate indicator is used on each working cylinder of a multi-cylinder engine (each end).

18. Brakes or absorption dynamometers when used must be suitable for the engine to be tested, in capacity, in speed-torque relations, and in ability to maintain an adjustment without necessity for constant hand manipulation to hold scale beam steady.

(*a*) For testing all engines of Par. 5(*c*) and (*d*), nothing but an electric dynamometer shall be used to determine brake horsepower

(*b*) For brake-horsepower tests of engines of Par. 5(*b*), the electric dynamometer shall be used, or hydraulic, if reversing tests are to be run. On engines of this class of large size and lower speed, hydraulic brakes of the type in which brake torque varies with the nth power of the speed, and in which n is equal to or greater than 1.5, shall be used as the preferred form.

(*c*) Other forms of brake may be properly used with engines of Par. 5(*a*) and so also when available may the hydraulic type and the electric dynamometers above described.

(*d*) When brakes of any kind cannot be mounted directly on the engine shaft without other supports, due precautions must be taken, per instructions in the Code on Instruments and Apparatus, to avoid errors due to these supports.

(*e*) In no case may a brake driven by a belt or otherwise than by direct coupling to or mounting on the engine shaft, be used for engine brake-horsepower measurements.

(*f*) With due precautions and under proper provisions brake-horsepower determinations may be made by the indirect or substitution method, and this is important because in some cases it is the only means available. According to this method the engine may be tested with a load such as a propeller (air or water type), a club or a fan, which propeller or fan may later be, or previously have been, tested for its speed-torque characteristic curve, and the horsepower of the engine determined by reading from this curve the horsepower for the speed maintained by the engine. The special precautions referred to relate to assurances of identity of conditions of flow through and resistance of the propeller or the fan under the two separate tests of use and of calibration. This is so difficult that none but test engineers of special experience should undertake it. In no case can results of test-stand torque alone be acceptable in brake-horsepower determinations for an engine.

19. In measuring speed, the choice of instruments is determined by the character of the test carried out. Continuous speed counters may be used only for slow-running engines, where a constant speed of considerable duration is maintained. Hand speed counters placed directly against the engine shaft may be used, if handled carefully and well made. Tachometers may be used of the centrifugal, liquid pumping, or magnetic type, but these must be carefully calibrated and records of recent calibration must be available. Tachometers must be connected by a reliable tachometer shaft or by positive gearing to the engine shaft, and in no case should belting be used as a connecting link.

20. The heating value of fuels to be used in calculations of results of tests of internal-combustion engines must be the "high value" products condensed or the direct reading of a water calorimeter properly used in accordance with the procedure specified by the Code on Fuels.

Accordingly the following recommendations are made:

(*a*) When gaseous fuel is burned in an engine under test, a standard form of gas calorimeter shall be used as one of the instruments of the test, and its determination shall be used as the heating value of the gas as supplied. Calculations of heating value from the gas analysis are not permissible according to the code, but with all details of apparatus, methods and physical constants involved may be made the subject of previous agreement.

(*b*) When liquid fuel is burned in an engine under test, its heating value shall be determined in a standard bomb calorimeter by a recognized physical or chemical laboratory, the selection of which should be a matter of previous agreement, and in the report the "high value" should be used in calculating final results. In no case shall such a determination be part of the duty of a test engineer, nor should his results be regarded as acceptable.

(*c*) The heating value of gasolines and kerosenes may be estimated by the Sherman & Kropff formula, modified by Strong, from the Baumé reading without the bomb test, if such procedure is made the subject of previous agreement.

21. Gas measurements for engine burning gaseous fuel must be made with meters or by methods yielding the greatest possible accuracy consistent with availability and cost in the large sizes.

In no case can the results of proportional meters be regarded as satisfactory, nor those of vane meters of the anemometer type Within the limits of size and cost, positive displacement meters are to be preferred, but these are not available for measurements of large capacity

Large capacity measurements are best made by means of one of the various types of flow-rate measurements, and that one, applied to mixed gases, should be used that requires the simplest and most accurate determination of physical or chemical properties

When physical or chemical properties of a mixed gas must be determined for use in a metering calculation, even if only gas density direct or computed from the analysis, a recognized chemical laboratory shall be agreed upon to make the determination and its report shall be accepted and used. In no case shall the test engineer make a determination of physical or chemical properties of gases, nor shall his results be acceptable if he does so.

If measurements of volume of injection air or of scavenging air are desired, the orifice-flow method will be sufficiently correct if used as prescribed in the Code on Instruments and Apparatus, but the formula and conditions for such test must be agreed upon in advance between the engineers.

22. Liquid fuel supplied to an engine should never be measured by volume; it must be weighed directly as used. Combinations of weighing and volume measurements are acceptable if the volume of the tank itself is not the measure but only a reservoir from which the engine draws a supply, starting with a predetermined level marked by a hook gage or on a glass, and a weighed quantity subsequently supplied to restore the original level. In this case, the weight is the measure and the result acceptable if proper pre-

cautions are used to limit the weight error of reading levels to something less than the error of the scale weighing.

In no case shall meters be used to measure liquid fuel supplied to an engine.

When volatile fuels are used, and especially when heavy liquid fuels containing light volatiles must be heated to make them flow, precautions must be taken to prevent losses of weighed fuel by evaporation, and the main supply must be protected against evaporation that may fractionate or concentrate it and change its quality with time.

PREPARATIONS

23. Before proceeding with a test, Pars 4 to 8 of the "General Instructions" should be reviewed. The dimensions and physical conditions of all parts of the engine should be determined and recorded. The same should be done for all parts external to the engine to be tested, the condition of which might effect the success of the test in many ways.

24. Things external to the engine should always be given attention first, and if on inspection and preliminary trial anything should be found that might impair the value of the test, it should be corrected before starting or even before the condition of the engine is examined. In view of the great variety of engines and operating arrangements, it is impossible to enumerate all of these things, but for their suggestive value a few are noted below:

(*a*) Jacket-water supply should be assured in proper quantity, of proper cleanliness as to freedom from sediment, non-corrosive, and of a kind that will not deposit scale in the jackets when its temperature is raised. It is necessary also to arrange for as nearly constant a supply temperature as possible to maintain a steady state as to temperature of heated engine parts. The exceptions to this are: air-cooled engines; engines with boiling hopper jackets or with thermo-siphon circulation to tanks, or with any sort of circulation of a fixed amount of water between jackets and a radiator or cooler. In these cases a proper supply of cooling water or air for the cooler or radiator must be assured. Engines operating with oil in jackets are included in the last group. Air-cooled engines must be assured of a supply of cooling air proper in amount, velocity and direction as well as temperature.

(*b*) Engines operating with electric ignition, especially in the larger sizes, may be dependent on an external supply of current for the ignition system, or especially in small engines be supplied from primary or storage batteries. Assurance of continued supply at proper voltage for the whole test must be provided, and the most reliable means is an alternate source.

(*c*) When liquid fuel is to be used, there should be a sufficiently large supply of the grade to be burned to last for the whole period of the test, and in the case of viscous or semi-viscous oils, which do not mix well without heating, special care should be taken to assure the mixing of new oil supplied to the tanks with that already in the tanks. This is especially important for that group of oil engines most sensitive to grade of oil.

(*d*) Lubricating oil in sufficient quantities to complete the test should also be available before starting. When the lubricating system includes a tank or sump, especially when it is of considerable capacity, this should be drained and new oil provided or new oil in sufficient quantities properly mixed with it to prevent any changes during the test, other than those normal and incident to its operation.

25. If the engine is by previous agreement to be tested "as is," or in whatever condition it may happen to be, there is nothing further to be done before starting the test except to determine all those dimensions required to identify the engine, or necessary for the calculation of results.

On the other hand, and normally, the engine should first be put in proper condition to perform in a manner proper and normal to it, so that the test results will be not only a measure of what it did do, but also of what it is capable of doing.

In the absence of any agreement to the contrary, it will always be assumed that before starting the test the engine will be put in proper condition so far as can be done by cleaning, by stoppage of all leaks and by adjustment, including replacement of small parts, such as spark plugs or spray valves. Some suggestions are given as a guide to these several groups of items, but these must not be regarded as a substitute for the judgment of the test engineer, who should in all cases record not only the condition in which the equipment was found, but also everything done to change the condition.

26. Dimensions and other data not necessary for calculating test results but useful to identify an engine or to increase the value of future comparative analyses may properly include some items of its commercial specifications not properly classifiable as dimensions. Among such items, including some that are necessary for calculating test result, are:

(*a*) Bore, stroke, number of working cylinders and in the case of two-cycle engines, the same information for scavenging pumps, with notes as to single- or double-acting arrangement for either or both and notes regarding injection air compressor if used

(*b*) Dimensions of other principal fixed and running parts, diameter and length of all important bearings and pins, including piston rods and tail rods.

(*c*) Dimensions of certain minor parts, fixed and running parts having important functions, such as valve lift, port area, area through mixing valves of gas-burning engines, carburetor choke area and nozzle sizes, injection-pump plunger bore and stroke, push-rod or valve-rocker clearances

(*d*) All items of adjustment and timing and for each cylinder of multi-cylinder engines such as spark of electric-ignition engines, spray valve of injection-oil engines when mechanically operated, opening and closing angles, injection pumps of injection engines delivering directly to spray valves not mechanically operated, start and stop angles, normal pump by-pass valve, timing of injection-oil engines, inlet and exhaust valves of four-cycle engines opening and closing, all special valves of two-cycle engines

(*e*) All name-plate information, together with important identification items of specifications especially concerning attached auxiliaries, nature, size, capacities, with proper working pressures, temperatures or other items of operation or adjustment

(*f*) Certain dimensions in addition to the above, not matters of identity nor necessary for calculation of test results but of importance in judging the condition of the engine as a machine and useful in judging the value of the results of the test. Here are included all items of alignment, clearances of bearings, pins, rods and slides, pistons in cylinders, rings in grooves, valve stems in guides, pump plungers in pump body, throttle in seat, together with variations from straightness or roundness of cylinders, pistons and other parts.

27. Cleanliness, especially internal, is of great importance in the operation of internal-combustion engines. Attention should be given, among other things, to the following suggestions:

(*a*) Jacket surfaces on the heat-receiving side must be free from water scale, rust or other deposits, including carbonized oil, if oil is used in place of water. Any deposit will cause the engine to operate unduly hot and may prevent operation entirely.

(*b*) Combustion-chamber walls, including piston heads and all inward-facing parts, such as spark plugs, spray valves, air-starting valves, relief valves, as well as inlet and exhaust valves and the ports of four-cycle engines, and the ports of two-cycle engines, should be free from deposits, especially carbon, oil, gum or tar, and solids derived from uncleaned air. This also applies to cylinder walls and piston rings.

(*c*) The lubricating-oil system should be free of all deposits of whatever kind, and all flow passages freely and fully open, especially with circulating forced-feed systems.

(*d*) The exhaust system, including expansion chambers, mufflers and pipes should be clean and free and so arranged as to not develop excessive back pressure. Other engines exhausting into the same system must not interfere with the engine under test. It is always preferable however that no other engines should exhaust into the system of the engine under test.

(*e*) Fuel-supply and regulating systems must be clean and free of sediment, tar or other foreign matter, and passages fully open, especially for liquid-fuel engines, at the carburetor or injection pumps.

28. Leakages must be located and corrected. Among important leakages to be investigated attention is called to the following:

(*a*) Cylinder leakage outward, lowering the compression and efficiency or possibly preventing operation. These leakages may be best checked by an indicator if the engine is one suitable for indicator use. When there is any clear space between the compression and expansion lines of a card taken when there is no combustion and when cooling water is not running then it may be assumed that leakage is excessive. In this case, and in cases where an indicator cannot be used to check leakage, the several possible sources of such leakage must be separately checked. These are piston ring, cylinder, cylinder-head gasket, air starting valve, relief valve, spray valve or spark-plug seat, inlet or exhaust valves of four-cycle engines, or air-scavenging valves of valve-scavenging two-cycle engines.

(*b*) Leakage into the cylinder may occur from the jackets through bad joints, cracks or porous spots and blowholes in the castings, compressed air from the starting air system through leaky air-starting valves, fuel oil past spray valve at improper times and high-pressure spray air when air spraying is used in oil engines.

(*c*) Miscellaneous leakages not into or out of the cylinder but elsewhere may be sought; in the manifold system of carburetor engines anywhere between carburetor and inlet-valve seat, especially at flanges and valve-stem guides; fuel-oil in injection-oil engines at pump valves, pump plunger, oil-delivery pipe or spray valve, lubricating oil at pump, tanks or in piping, spray air at compressor storage bottles, valves or piping, starting air at compressor storage reservoir, valves or piping. Insulation of electrical circuits must be checked as a leakage item.

29. Adjustments must be corrected if found to be wrong.

OPERATING CONDITIONS

30. The operating conditions conforming to the object in view should prevail as pointed out in Par. 19 of the "General Instructions." If it is a matter of agreement that the engine is to be tested in whatever condition it may be, and under whatever conditions may prevail, then there is nothing to be done except to keep a record of all these conditions, with special emphasis on variations of every item that might effect the operation, or be useful in a subsequent effort to explain results.

On the other hand, if by agreement operating conditions are specified more or less completely, then instruments shall be used and observations made to establish their maintenance or variations.

In either case a record should be made and included in the final report.

This matter of operating conditions prevailing during the test is of very great importance in internal-combustion engines, not only because of their great variety and the corresponding variety of conditions normal to any one sort of engine, but also because the internal-combustion engine is self-contained, actually or essentially, and sensitive to many conditions within itself, not noticeable unless sought. Many of these have already been noted under "Preparations," as concerned with supplies, flows, leakages, clearances and fits or adjustments, but in addition to these there always arise questions of propriety of conditions of test in relation to condition of service.

Doubt and confusion will always result in internal-combustion engine tests if any assumption is made to the effect that the conditions of test shall be "normal." If the word "normal" is used at all to describe the conditions, it must be defined by agreement. An agreement defining the conditions of the test is regarded as essential to all internal-combustion engine tests. The definition should include all the items necessary in two groups noted below, one concerned with conditions in relation to service, and the other purely with matters of test procedure.

31. Operating conditions to be maintained during the test are the ones most uncertain as will be clear from the following note:

Aircraft engines must normally work in altitudes varying from sea level to perhaps 30,000 ft., and in temperatures varying from highest in summer in the tropics, perhaps 120 deg. F., to the coldest known, under 50 deg F. below zero, and in any weather—dry, fog clouds, rain and snow. They must also operate well at any angle of inclination and even upside down but always with an air-propeller load. In view of the fact that the only tests contemplated under this code are fixed tests, it is clear that service conditions cannot be closely reproduced and only approximated even in the special altitude room of the Bureau of Standards with its air-refrigerating apparatus.

Engines for land vehicles are essentially variable in speed and variable in load in normal operation, and must instantly respond to controls that are constantly and suddenly varied in heavy street traffic, or perhaps maintained for long periods of time in tractor or rail operation, and rarely develop maximum horsepower though frequently called upon for maximum torque. There can be no such thing as normal horsepower or speed for such engines and acceleration is quite as important as horsepower. They must, furthermore, operate on a fixed supply of jacket water or oil with an air-cooled radiator or directly air-cooled, so that external wall temperature varies not only with engine load but with vehicle speed, with weather and with climate.

As used, especially with tractors, the air supply is usually dirty, and may be heavily laden with gritty dust, requiring the use of more or less effective dust separators or air washers. Again, it is clear that service conditions cannot be maintained during the block tests contemplated by this code.

Marine engines, ranging from small single- and multiple-cylinder gasoline engines, often of very high speed, especially for racing boats with gear drives and clutches, to the direct-coupled large reversing low-speed Diesel oil engine of motorships of 2,000 to 3,000 hp., always with a propeller load, also operate under service conditions that cannot be reproduced in block tests.

Even stationary engines as a class while normally operating under conditions easier to reproduce in the block tests or tests in place according to this code, do also include considerable variety as to what is normal. Some of these, especially the smaller ones, such as farm engines or home-lighting sets, operate for only short periods of time, and rarely if ever at full load. Others, especially those for irrigation pumping, may work continuously at the maximum possible load 24 hours per day for the irrigation period, of perhaps three months, and then be shut down for the rest of the year. Still others, especially those operating town water works or oil-pipe-line pumps may operate at full load 24 hours per day for a whole year or more Still others, notably the generating units of central stations, must meet the widest and most sudden fluctuation of electrical load without varying in speed enough to affect voltage or synchronous operation of alternating-current systems.

The importance of agreement on these matters should be clear from these citations but the details must be left to the judgment of the test engineer.

32. Operating conditions during the test include mainly matters of load and speed to be maintained, with other points incidental to these. These should be agreed upon in all details, and should not be inconsistent with the conditions prescribed below:

(*a*) Engines of the stationary class, including all engine-driven generator sets, Par. 5(*a*), shall be tested at constant speed under governor control or as near constant as the governor will maintain it, and at whatever load in horsepower or kilowatts may be required by agreement or specific object of the test.

(*b*) All engines for the propulsion of land vehicles, Par. 5(*d*), except such tractor engines as are fitted with speed governors and operate normally under governor control, shall be tested with throttle wide open, with brake torque regularly varied from nothing to a maximum and back to nothing, at whatever speeds may result, and the horsepower-speed curve typical of the engine shall be determined instead of the horsepower at any given speed. In no case should a *safe* speed be exceeded and when the highest speed used is the safe limit, it should be so stated. The maximum torque applied shall in all cases be greater than that which will produce maximum horsepower unless the speed required for this is too high to be safe.

(*c*) Marine engines, Par. 5(*b*), and aircraft engines, Par. 5(*c*), may be tested according to the conditions prescribed for engines of land vehicles in all respects according to conditions prescribed for stationary engines except that the speed shall be maintained by the brake torque and not by a governor. Marine engines may also be tested at constant speed for full-load operation fixed by their propellers

Special attention should be given to questions of safety that may arise during operation, such as excess speed, or torsional vibrations at a given speed within the normal range, and the procedure to be adopted in such cases should also be included in the agreement.

STARTING, STOPPING AND DURATION

33. Before recording the test observations it is essential that an internal-combustion engine, together with all attachments and appurtenances, be brought to a condition of *steady state.* The test cannot be considered as having started until the engine has been in operation for a sufficiently long time to have attained its steady state for the conditions of the first run, and until preliminary observations have been made and recorded to prove that such steady state has been reached. If successive runs are to be made, each under some different conditions, these must be repeated for each run.

The preliminary observations establishing the attainment of a steady state for the conditions of operation of the test or each run of the test, must be made a part of the record.

For an internal-combustion engine itself this is a matter of utmost importance, because the combustion chamber is a furnace in which heat is developed at a given rate, depending on the fuel burned per minute in each cylinder (or each end), and in which the metal temperatures vary with this rate of combustion. Metal temperatures vitally affect the performance of the engine especially those of cylinder, cylinder head, and piston, with all attached parts, and in addition those of all bearings.

The length of time necessary to establish the steady state will be different in engines of different class, and in any one class different sizes. In general, the weight of metal in contact with the combustion chamber directly or indirectly through thermal conduction will determine the length of time necessary, but the rate of combustion in British thermal units per hour per square foot of cylinder area will also be a factor. The greater the metal weight and the lower the rate of combustion the larger the time necessary to reach the steady state. While jacket-water or oil temperatures are important factors, they are not the controlling ones. Lubricating-oil temperature, in the sump or leaving the bearings in a circulating system, is a guide to steady state of both bearings and of combustion-chamber metal, but not a conclusive one. In no case shall an engine be regarded as having attained its steady state unless jacket and lubricating-oil temperatures are substantially constant.

The means whereby the attainment of the steady state is proved must be a matter of agreement, but the time of operation itself for the preliminary period may be substituted.

In no case shall steady state for the first run be regarded as established in a shorter time than one hour after the operating conditions have been imposed and before test observations have started. For successive runs at other conditions the minimum time for any engine shall be 10 minutes, the actual time being determined by the extent to which conditions are changed in successive run. Maximum time to reach the steady state for any internal-combustion engine shall not be taken as greater than 24 hours, which may be regarded as permissible for the largest sizes of low-speed Diesel engines for motorships and double-acting stationary gas engines.

34. The duration of the test after the establishment of steady state for set operating conditions shall be a matter of previous agreement, but should be greater for engines where reliability is a question of importance and which require the longest time to reach the steady state.

In no case shall the length of run be less than the period required to reach the steady state, subject to the additional condition that the length of run shall be great enough to insure the accuracy of fuel measurement within 1 per cent, except when gaseous fuel is metered and when metering accuracy is not improved by lengthened runs.

In some cases it may be found desirable to make a physical inspection of engine parts after the test to determine injury due to operation, but only when agreement has been made as to details should this be done.

RECORDS

35. The general data should be recorded as pointed out in Pars. 20 to 30 of the "General Instructions." Instruments should be read at least quarter-hourly when the conditions are uniform, and oftener when there is much variation. Indicator cards, if taken, must be taken from every working cylinder of multi-cylinder engines. If there are wide fluctuations in readings, they should be shown by recording instruments.

Each indicator card should be marked when taken, with the number, date, time, scale of spring, number and kind of cylinder and end of cylinder, if double-acting.

The log should contain the records of the readings of all instruments, and these readings should be obtained at practically the same time as the indicator cards, if any are taken.

If indicator cards are taken, the areas, lengths, mean effective pressures, compression pressures and maximum cylinder pressures, should all be recorded in the log. One or more sets of specimen indicator diagrams should be carefully selected for inclusion in the record. If, after taking indicator cards they are rejected, because of excessive fluctuation in area or insufficient area with high pressures for accuracy, a sample set of the rejected cards shall be included in the record with the data to support the rejection and justify the elimination of indicated horsepower from the results.

CALCULATION OF RESULTS

36. The calculations necessary for deriving results from observations or from physical constants should not be undertaken without previously consulting Pars. 31 to 35 of the "General Instructions," and the Code on Definitions and Values.

37. *Fuel Consumption.*—For liquid-fuel engines the actual fuel measured is stated in pounds in the report of the test, and is subject to no corrections.

38. *Heat Consumption.*—The number of heat units consumed by an engine per hour is found by multiplying the consumption by the heating value. The heating value for gaseous fuels is stated in the report of the test as found by the gas calorimeter, in B.t.u. per cubic foot, high value. The heating value of liquid fuels is stated in the report of the physical or chemical laboratory engaged for the determination, in B.t.u. per pound of fuel, high value.

39. *Indicated Horsepower.*—If indicated-horsepower determinations have been specified in the agreement before test and when the indicator can be used, and when in addition after using it the indicator cards are found to be acceptable as to accuracy for determining mean effective pressure, per Par. 16, then the indicated horsepower shall be calculated as follows:

For four-cycle engines the indicated horsepower for each single-acting cylinder and for each end of each double-acting cylinder shall be found by using the formula—

$$\text{I.hp.} = \frac{PLAN}{2 \times 33{,}000}$$

where P represents the mean effective pressure in pounds per square inch; L the length of the stroke in feet; A the area in square inches of the piston less the area of the piston rod or tail rod, if any; and N the number of revolutions per minute. The total indicated horsepower of a double-acting cylinder is the sum of the horsepower developed in the two ends, and the indicated horsepower of a multi-cylinder engine is the sum of the horsepower developed in each cylinder.

For two-cycle engines the indicated horsepower for each single-acting working cylinder, and for each end of each double-acting working cylinder, shall be found by using the formula—

$$\text{I.hp.} = \frac{PLAN}{33{,}000}$$

where the symbols have the same meaning as above. The total indicated horsepower of all working cylinders is the sum of the horsepower developed in each cylinder, and in each double-acting cylinder it is the sum of the horsepower developed in each end.

The mean effective pressure, P, should be found by dividing the area of the diagram in square inches, as determined with a correct planimeter, by the length of the diagram, in inches, and multiplying the quotient by the average corrected scale of the indicator spring.

Accuracy tests must be applied to determine the acceptability of the mean effective pressure thus determined, especially when cylinder pressures and spring scales are high (as for injection-oil engines) and card areas small, as is more usual for two-cycle engines of smaller size.

40. *Brake Horsepower.*—The brake horsepower is found by multiplying the net force, W in pounds on the brake arm, by the circumference of the circle whose radius is the horizontal distance, L, in feet between the center of the shaft and the bearing point or weight center at the end of the brake

arm, and by N, the number of revolutions of the brake shaft per minute, and dividing the final product by 33,000.

$$\text{B.hp.} = \frac{2\pi LWN}{33{,}000}$$

Reference must be made to the Code on Instruments and Apparatus for descriptions of brakes, especially the hydraulic and electric dynamometer types, referred to in Par. 17 for methods of applying them, and especially for methods of determining correctly the net force W from the gross force, and for incidental or special errors in brake use and calculations.

41. *Electrical Horsepower.*—The electrical horsepower of a generator is found by dividing the output at the terminals expressed in kilowatts by the constant 0.7457. In the case of an alternating-current generator, the quantity of output determined whether expressed in electrical horsepower or kilowatts, should be the net output. When the power for excitation or ventilation is taken directly from the engine shaft, the net output is that indicated at the terminals. When the exciting current is obtained from the generator through a motor or from some outside source, the net output is found by deducting the current furnished as excitation from that delivered at the terminals. When the exciter is driven from the engine shaft, the engine supplies the belt losses and the exciter losses without being credited. When the exciting current is obtained from the generator through a motor, and the excitation current is deducted from that delivered at the terminals, the exciter losses are not credited. Whereas with a separately driven exciter, it is really the exciter net output which is deducted from the power delivered at the terminals. In a close test these differences may be of some interest. It is regarded as important to have the basis of power determination definitely agreed to in advance to suit the particular case and to be suitable to local conditions prevailing.

42. *Thermal Efficiency.*—The fraction of the heat consumption converted into work is the "thermal efficiency" and is found by dividing 2,547 (B.t.u. equivalent to 1 hp.-hr.) by the number of heat units actually supplied per indicated horsepower hour, Par. 37. The quotient is multiplied by 100 to express the thermal efficiency in per cent. The formula is

$$\text{Thermal Efficiency} = \frac{2{,}547}{HQ}$$

where H = heating value of the fuel (high value), British thermal unit per pound or per cubic foot and Q = amount of fuel supplied per indicated horsepower hour, pounds or cubic feet.

Data and Results

43. The data and results should be reported in accordance with the form of Table 1 for horsepower tests made at constant speed, and in accordance with the form of Table 2 and chart, Fig. 1, for horsepower tests made over the whole range of a variable-speed engine. Lines may be omitted if not required for the object of the test, or new lines may be added for additional data desired. For marine engines either or both forms may be used.

TABLE 1.—DATA AND RESULTS OF INTERNAL-COMBUSTION ENGINE TEST AT CONSTANT SPEED

GENERAL INFORMATION

(1) Date of test....... ... .
(2) Location ....
(3) Owner......... ....
(4) Builder.. ... .
(5) Test conducted by . . .. .
(6) Object of test . . .

DESCRIPTION, DIMENSIONS, ETC.

(7) Type of engine, two- or four-cycle, single- or double-acting, horizontal or vertical, if four-cycle, the valve arrangement L-, T-, or I-head, if two-cycle, the type of scavenging, if single-acting, whether crosshead or trunk piston, if multi-cylinder the arrangement of cylinders and cranks, gas or liquid fuel, and if liquid fuel, carburetor type, or other class name fixing manner of treating fuel, such as Hvid, Diesel, or solid injection for example ....
....

(8) Class of service, stationary and special feature, marine, aircraft or vehicle, and what kind . ...

(9) Auxiliaries attached, such as magneto, fuel-injection pump, fuel-circulating pump, lubricating-oil pumps, jacket-circulating pumps, scavenging pumps, spray-air compressor, maneuvering or starting air compressor, radiator fans, oil or fuel coolers or heaters . . . . .

(10) Auxiliaries, independent or separately driven, and power . .

(11) Details of type, capacity, maker's name and other features of auxiliaries.................. . ...

(12) Rated brake horsepower of engine, or kw. of electric generating set, and speed. .. ..

(13) Grade of fuel for which engine is designed, kind of gas or specification of liquid fuel, and what was used in test ... ...

(14) Special structural features for fuel utilization . .. .

(15) Special structural features of speed and power control, and governor or reversing gear....... ....

Number of working cylinders ....

(16) Diameter of all cylinders working, four-cycle or two-cycle ..... in.

(17) Diameter of piston and tail rods of working cylinders ....in.

(18) Stroke of working cylinders.... ....ft.

(19) Head-end hp. constant for working cylinders (stroke × net piston area ÷ 33,000).................... ... .

(20) Crank-end hp. constant for working cylinders (stroke × net piston area ÷ 33,000) ....

(21) Capacity of generator or other apparatus consuming power of engine.

(22) Characteristics of generator—d.c. or a.c., volts, cycles, phase .. ...

Test Data and Results

(23) Duration of test..hr.

Pressures, Average

(24) Barometric pressure......in. of mercury......lb. per sq. in.

(25) Spray-air pressure (air injection Diesel engines) average (say whether gage or absolute)..................................lb. per sq. in.

(26) Exhaust back pressure............................in. of water

(27) Jacket-water supply pressure.....................lb. per. sq. in.

(28) Manifold vacuum (carburetor engines)...............in. of water

(29) Gas pressure in main gaseous fuel.......in. water..lb. per sq. in.

(30) Lubricating-oil pressure, circulating forced-feed system.lb. per sq. in

(31) Scavenging-air pressure average, two-cycle engines..lb. per sq. in.

Temperatures

(32) Engine-room temperature..............................deg. F.

(33) Temperature of fuel...................................deg. F.

(34) Temperature of main air supply..........................deg. F.

Location of thermometer...

(35) Temperature of main jacket-water or oil inlet..............deg. F.

(36) Temperature of main jacket-water or oil outlet............ deg. F.

(37) Temperature at inlet of separate jacket...................deg. F.

(38) Temperature at outlet of separate jacket..................deg. F.

(39) Temperature of mixture at intake port (carburetor engine)...deg. F.

Fuel Properties

(40) Heating value of gas, high value..............B.t.u. per cu. ft.

(41) Heating value of liquid fuel, high value ....B.t.u. per lb.

Total Quantities

(42) Total gaseous fuel at meter pressure and temperature......cu. ft.

(43) Correction factor for gas (Item 29 × Item 33 + 460 ÷ Item 29 × Item 39 + 460)..

(44) Total gaseous fuel at carburetor or atmospheric pressure..cu. ft.

(45) Total liquid fuel..lb.

Hourly Quantities

(46) Total gaseous fuel per hour at calorimeter or atmospheric pressure and temperature..cu. ft.

(47) Total liquid fuel per hour................................lb.

Heat Consumption

(48) Total heat in fuel supplied per hour (Item 46 × Item 40, or Item 47 × Item 41)..B.t.u.

Indicator Diagrams

(49) Mean effective pressure, average, four-cycle....lb. per sq. in.

(50) Mean effective pressure working cylinders, average, two-cycle...................................lb. per sq. in.

Speed

(51) Revolutions per minute...............................r.p.m.

(52) Piston speed..................................... ft. per min.

Power

(53) Indicated horsepower of all working cylinders............i.hp.

(54) Indicated horsepower of each single acting cylinder, or each end of each double acting four cycle

No. 1 head end.. . . . . .i.hp.
No. 1 crank end. . . . .i.hp.
No. 2 head end.. . .i.hp.
No. 3 crank end . . . . .i.hp.
etc.

(55) Brake horsepower developed by whole engine by brake or dynamometer measurement.......................... . ...b.hp.
(*a*) Brake mean effective pressure............ .. ..lb. per sq. in.
(56) Total horsepower of independent scavenging pump and injection air compressor............. ....hp.
(57) Net or actual brake horsepower of engine,
(Item 55–Item 56) ....b.hp.
(*a*) Net brake mean effective pressure .. ..lb. per sq. in.
(*b*) Net torque at 1 ft. radius equivalent to brake horsepower....ft.-lb.
(58) Friction and attached auxiliaries horsepower (Item 53-Item 55) .hp.

Economy Results

Fuel consumption per i.hp.-hr., gas (Item 46 ÷ Item 53) .. cu. ft.
Fuel consumption per i.hp-hr., liquid (Item 47 ÷ Item 53) ....lb.
(59) Fuel consumption per b.hp-hr.: Gas (Item 46 ÷ Item 57) cu. ft.
Liquid (Item 47 ÷ Item 57)........................ . .. .lb.
(60) Heat consumed per i.hp-hr.: (Item 48 ÷ Item 53) ... ...B.t.u.
(61) Heat consumed per b.hp-hr. (Item 48 ÷ Item 57) . . . B.t.u.

Efficiency Results

(62) Thermal efficiency referred to i.hp. (2,547 ÷ Item 60) .. per cent
(63) Thermal efficiency referred to b.hp. (2,547 ÷ Item 61) . per cent

Specimen Diagrams

(64) Sample diagram each working cylinder.. . .. .

Electrical Data

(65) Average volts each phase ....volts
(66) Average amperes each phase......... .... amp.
(67) Total electrical output corrected for winding 1, 2 or 3 phase kva.
(68) Power factor.................... .... per cent
(69) Total output in kilowatts.................. ....kw.
(70) Field power.................. .. ..kw.
(71) Net output in kilowatts ....kw.
(*a*) Field volts.............. . ...
(*b*) Field amperes........... ....

Power and Economy

(72) Electrical horsepower developed (Item 70 ÷ 0.7457) ....
(73) Fuel consumed per net kw.-hr. Gaseous (Item 46 ÷ Item 70). cu. ft.
Liquid (Item 53 ÷ Item 83).............. ....lb.
(74) Heat consumed per net kw.-hr. (Item 48 ÷ Item 70).....B.t.u.

TABLE 2.—DATA AND RESULTS OF INTERNAL COMBUSTION ENGINE TEST AT VARIABLE TORQUE AND SPEED

GENERAL INFORMATION

(1) Date of test.................................... . ...
(2) Location.................................... ... .

(3) Owner

(4) Builder

(5) Test conducted by

(6) Object of test

DESCRIPTION, DIMENSIONS, ETC.

(7) Type of engine, carburetor or oil, single- or double-acting, if four-cycle, the valve arrangement L-, T-, or I-head, if two-cycle the type of scavenging, if single-acting, whether crosshead or trunk piston, if multi-cylinder, the cylinder and crank arrangement and if radial cylinders whether cylinders are fixed or rotate, and, means of fuel utilization by class name, such as Hvid, Diesel, or solid injection for example

(8) Class of service, aircraft, automobile, truck, tractor, railroad or marine, for gear, electric or direct-propeller drive, single direction or reversing

(9) Auxiliaries, attached and independent, kind, maker's name, capacity, etc

(10) Rated horsepower and speed, if any, or speed at maximum torque

(11) Bore and stroke of all working cylinders

(12) Weight of engine complete

		Run Number 1	2	3
(13) Brake horsepower	b.hp.			
(14) Speed	r.p.m.			
(15) Total fuel	lb.			
(16) Time of run	hr.			
(17) Fuel per hour	lb.			
(18) Fuel per hour per b.hp.	lb.			
(19) Heating value of fuel ... B.t.u.	lb.			
(20) Heat consumed per hour (Item 17 × Item 19)	B.t.u.			
(21) Heat consumed per hour, per b.hp. (Item 20 ÷ Item 13)	B.t.u.			
(22) Thermal efficiency (2,547 ÷ Item 21)	per cent			
(23) Mean effective pressure equivalent to brake horsepower	lb. sq. in.			
(24) Torque at one foot radius equivalent to brake horsepower	ft.-lb.			
(25) Fig. 1 Curve of r.p.m. plotted horizontal against vertical				
(*a*) Brake horsepower, Item **13**				
(*b*) Fuel per hour (Item 17)				
(*c*) Fuel per hour per b.hp. (Item 18)				
(*d*) Thermal efficiency (Item 22)				
(*e*) Brake mean effective pressure (Item 23)				
(*f*) Torque (Item 24)				

CHAPTER XXIV

MARINE RULES

General.—It has been thought advisable to add the rules issued by Lloyd's and the Bureau of Commerce.

LLOYD'S RULES

Section 1.—In vessels propelled by Diesel-Oil Engines, the Rules as regards machinery will be the same as those relating to steam engines, as far as regards the testing of material used in their construction and the fitting of sea connections, discharge pipes, shafting, stern tubes, and propellers

CONSTRUCTION

Section 2.—In vessels built under Special Survey and fitted with Diesel Engines, the engines must also be constructed under Special Survey.

2. In cases of Diesel Engines being built under Special Survey, the distinguishing mark ✠ will be noted in Red, thus: ✠LMC or ✠NE.

3. In order to facilitate inspection, the plans of the machinery are to be examined by the Surveyors, and the dimensions of the shafts are to be submitted for approval.

4. The Surveyors are to examine the materials and workmanship from the commencement of the work until the final test of the machinery under full working conditions; any defects are to be pointed out as early as possible.

5. Any novelty in the construction of the machinery is to be reported to the Committee and submitted for approval.

6. The auxiliary engines used for air compressing, working dynamos and ballast, or other, pumps, are also to be surveyed during construction.

7. In cases where the designed maximum pressure in the cylinders does not exceed 500 lb. per square inch, the diameters of the crankshaft of the main engines are not to be less than those given by the following formula:

$$\left.\begin{matrix}\text{Diameter of}\\ \text{crankshaft}\end{matrix}\right\} = \sqrt[3]{D^2 \times (AS + BL)}$$

where D = diameter of cylinder,
S = length of stroke,
L = span of bearings adjacent to crank, measured from inner edge to inner edge.

The value of $(AS + BL)$ are as given in the following table:

TABLE I

Four-cycle single acting engine, cylinders	Two-cycle single acting engine, cylinders	Values of the coefficients
4 or 6	2 or 3	$0.089S + 0.056L$
8	4	$0.099S + 0.054L$
10 or 12	5 or 6	$0.111S + 0.052L$
16	8	$0.131S + 0.050L$

For auxiliary engines of the Diesel Type the diameters of the crankshafts may be 5 per cent less than given by the foregoing formula.

8. In solid forged shafts the breadth of the webs should be not less than 1.33 times and the thickness not less than 0.56 times the diameter of the shaft as found above, or, if these proportions are departed from the webs must be of equivalent strength.

9. The diameter of the intermediate shaft must not be less than that given by the formula:

$$\left.\begin{array}{l}\text{Dimeter of inter-}\\ \text{mediate shaft...}\end{array}\right\} = C\sqrt[3]{D^2 \times S}$$

where D = the diameter of cylinder,

S = the stroke of piston,

C is a coefficient found from the following table by interpolation from the values found for A.

Where the stroke is not less than 1.2 times, nor more than 1.6 times the diameter of the cylinder, $(0.735D + 0.273S)$ may be taken instead of $\sqrt[3]{D^2 \times S}$.

TABLE II

Two-cycle single acting engines, cylinders	Values of the coefficient *C* where $A = 0.0025$	$A = 0.0050$	$A = 0.0100$
2	0.305	0.317	0.336
3	0.346	0.363	0.385
4	0.364	0.380	0.396
5	0.380	0.391	0.404
6	0.398	0.403	0.412

Four-cycle single acting engines, cylinders	Values of the coefficient *C* where $A = 0.0025$	$A = 0.0050$	$A = 0.0100$
4	0.300	0.312	0.327
6	0.338	0.355	0.370
8	0.357	0.366	0.376
10	0.376	0.382	0.389
12	0.394	0.398	0.404

In using the above table the appropriate value of *A* is found from

$$\mathbf{A \times W \times d^2 \times R^2 = D^2 \times S}$$

where **D** = diameter of cylinder in inches,
S = stroke of piston in inches,
d = diameter of flywheel in feet,
R = revolutions of engines per minute,
W = total weight of flywheel in tons.

10. The diameter of the flywheel shaft must be at least equal to that of the crank shaft.

11. Where ordinary deep collars are used the diameter of the thrust shaft measured under the collars must be at least 21/20ths that of the intermediate shaft. The diameter may be tapered off at each end to the same size as that of the intermediate shaft.

12. The diameter of the screw shaft must be not less than the diameter of the intermediate shaft (found as above) multiplied by $\left(0.63 + \frac{0.03\mathbf{P}}{\mathbf{T}}\right)$ but in no case must it be less than 1.077,

where **P** = the diameter of the propeller in inches,
T = the diameter of intermediate shaft in inches.

The size of the screw shaft is intended to apply to shafts fitted with continuous liners the whole length of the stern tube, as provided for in Section 11, paragraph 3, of the Rules for Engines and Boilers for Steam Vessels. If no liners are used, or if two separate liners are used, the diameter of the screw shaft should be 21/20ths that given above.

The diameter of the screw shaft is to be tapered off at the forward end to the size of the thrust shaft.

13. If the designed maximum pressure in the cylinders exceeds 500 lb. per square inch, the diameters of the shafting throughout must be increased in the proportion of $\sqrt[3]{\frac{\text{Maxim. press. in lbs. per sq. in.}}{500}}$

14. Where the cylinder liners are made of hard close-grained cast iron of plain cylindrical form, accurately turned on the outside as well as bored on the inside so that their soundness can be ascertained by inspection, and their thickness at the upper part is not less than $\frac{1}{15}$th of the diameter of the cylinder, they need not be hydraulically tested by internal pressure. If, however, they are made of complicated form, the question of testing must be submitted.

15. The water jackets of the cylinders, and the water passages of the cylinder covers and pistons, must be tested by hydraulic pressure to 30 lb. per square inch, and must be perfectly tight at that pressure.

16. The exhaust pipes and silencers must be water-cooled or lagged by non-conducting material, where risk of damage by heat is likely to occur.

17. The cylinders are to be fitted with safety valves loaded to not more than 40 per cent above the designed maximum pressure in the cylinders and discharging where no damage can occur.

18. The air compressors and their coolers are to be so made as to be easy of access for overhaul and adjustment.

19. Where the fuel is injected into the cylinders by air pressure, the following conditions are to be observed:

In single screw vessels, an auxiliary air compressor is to be provided of sufficient power to enable the main engines to be kept continuously at work when the main compressor is out of action.

If the maneuvering gear is arranged so that the engines can be kept continuously at work with some of the cylinders out of action, the auxiliary compressor need only be of sufficient power to enable the engines to be kept at work under these conditions.

In twin screw vessels in which two sets of compressors are fitted, the auxiliary compressor must be of such size as to enable it to take the place of either of the main compressors. If in such engines each main compressor is sufficiently large to supply both engines, a smaller auxiliary compressor will be sufficient.

20. A small auxiliary compressor, worked by a steam engine, or by an oil engine not requiring compressed air, is to be fitted for first charging the air receivers.

21. At least one high pressure air receiver is to be arranged with connections to enable it to be used for fuel injection, in case the working receiver of either main engine is out of use from any cause.

22. The circulating pump sea suction is to be provided with an efficient strainer which can be cleared inside the vessel.

23. In all vessels fitted with engines in which the lubricating oil is circulated under pressure a spare oil pump is to be supplied with all connections ready for immediate use, and two independent means are to be arranged for circulating water through the oil cooler.

Air Receivers and Pipes

Section 3.—(1) Compressed air receivers for starting air are to be supplied of sufficient capacity to permit of twelve consecutive startings of the engines without replenishment.

2. Cylindrical receivers for containing air under high pressure, used either for starting or for the injection of fuel in oil engines, may be made either of seamless steel or of welded, or riveted, steel plates.

3. **Quality of Metal.**—If made of welded, or riveted, steel plates, the ordinary rules regarding steel material for boilers apply, which provide that where welding is employed, either in the longitudinal seams or at the ends, the material must have a tensile strength not exceeding 30 tons per square inch (Section 33, Par. 7, Rules for Engines and Boilers). In these cases the welding must be lap welding; neither oxy-acetylene nor electric welding will be permitted.

4. In the case of seamless receivers, the rules for material will be the same as for boiler shells, but the permissible extension may be 2 per cent less than that required with boiler plates.

5. **Tensile and bend tests** are to be made from the material of **each** receiver. When they are welded or riveted, the tests may be made, and the thicknesses verified before the plates are bent into cylindrical form. In the cases of seamless receivers, the thicknesses must be verified by the Surveyor before the ends are closed in, and at this time the Surveyor shall

select and mark the test pieces required from either of the open ends of the tube. The test pieces are to be annealed before test, so as to properly represent the finished material.

6. The permissible working pressure for welded or seamless receivers is to be determined by the following formula

Maximum working pressure in pound per square inch

$$= \frac{C \times S \times (T - 2)}{D}$$

for thicknesses of $\frac{5}{8}$ in. and above,

$$= \frac{C \times S \times (T - 1)}{D}$$

for thicknesses below $\frac{5}{8}$ in.,

where **S** = Minimum tensile strength of the steel material used, in
T = Thickness of the material, in sixteenths of an inch.
D = Internal diameter of cylinder, in inches,
C = Coefficient as per following table

Coefficient—
77 for seamless receivers of thickness of $\frac{5}{8}$ in. and above.
69 for seamless receivers of thickness below $\frac{5}{8}$ in.
51 for welded receivers of thickness of $\frac{5}{8}$ in. and above.
48 for welded receivers of thickness below $\frac{5}{8}$ in.

7. For flat ends welded into the cylindrical shells, the thickness must not be less than

$$T = \frac{D}{17} \times \sqrt[3]{P}$$

where **T** = thickness, in sixteenths of an inch,
D = internal diameter, in inches,
P = working pressure, in pound per square inch.

8. The permissible working pressure for receivers made of riveted steel plates is to be determined by the rules regulating the working pressure of boilers.

9. Each welded or seamless receiver shall be carefully annealed after manufacture, and before the hydraulic test.

10. Each welded or seamless receiver shall be subjected to a hydraulic test of twice the working pressure, which it shall withstand without permanent set.

11. Each receiver made of riveted steel plates for pressures up to 300 lb. per square inch is to be tested by hydraulic pressure 1½ times the working pressure, plus 50 lb. per square inch. Where higher working pressures are used, the test pressure need not be more than 200 lb. per square inch above the working pressure.

12. All receivers above 6 in. internal diameter must be so made that the internal surfaces may be examined, and, wherever practicable, the openings for this purpose should be sufficiently large for access. Means must be provided for cleaning the inner surfaces by steam, or otherwise.

13. Each receiver which can be isolated must have a safety valve fitted, adjusted to the maximum working pressure. If, however, the air compressor is fitted with a safety valve so arranged and adjusted that no greater pressure

than that permitted can be admitted to the receivers, they need not be fitted with safety valves.

14. Each receiver must be fitted with a drain arrangement at its lowest part, permitting oil and condensed water to be blown out.

15. Oil or air pipes subjected to high pressure are to comply with the Rules for steam pipes, Section 13, Clauses 7 and 16 (Rules for Engines and Boilers of Steam Vessels).

Pipes which are subjected to a working pressure up to 1,000 lb. per square inch must be tested hyraulically to at least twice the working pressure to which they will be subjected, and those subjected to a higher working pressure than 1,000 lb. per square inch to an hydraulic test of at least 1,000 lb. per square inch above their working pressure.

PUMPING ARRANGEMENTS

Section 4.—The pumping arrangements are to be the same as would be required for steam vessels of similar size and power, with the exception that no bilge suction need be fitted to the main engine circulating pump. In the cases of vessels fitted with water ballast, the water ballast pump must have, in addition, one direct suction from the engine room bilges.

GENERAL

Section 5.—(1) All oil fuel pipes, tanks and their fittings must comply with the requirements of Section 49 (Rules for Steel Ships).

2. Special attention must be given to the ventilation of the engine room.

3. If the auxiliaries are worked by electricity, the cables in connection with them must be in accordance with the rules for electric fittings.

SPARE GEAR

Section 6.—The articles mentioned in the following list will be required to be carried, viz.:

1 cylinder cover complete for the main engines, with all valves, valve seats, springs, etc., fitted to it.

In addition, one complete set of valves, valve seats, springs, etc., for one cylinder of the main and of the auxiliary Diesel engines, and fuel needle valves for half the number of cylinders of each engine.

1 piston complete, with all piston rings, studs, and nuts for the main engines.

In addition, one set of piston rings for one piston of the main and of the auxiliary Diesel engines.

1 complete set of main skew wheels for one main engine.

2 connecting rod, or piston-rod, top-end bolts and nuts, both for the main and for the auxiliary Diesel engines.

2 connecting-rod bottom end bolts and nuts, both for the main and for the auxiliary Diesel engines.

2 main bearing bolts and nuts, both for the main and for the auxiliary Diesel engines.

1 set of coupling bolts for the crankshaft.

1 set of coupling bolts for the intermediate shaft.

1 complete set of piston rings for each piston of the main and of the auxiliary compressors.

1 half set of valves for the main and for the auxiliary compressors.

1 fuel pump complete for the main engine, or a complete set of all the working parts.

1 fuel pump for the auxiliary Diesel engine, or a complete set of all working parts.

1 set of valves for the daily fuel supply pump.

1 set of valves for the water circulating pumps.

1 set of valves for one bilge pump.

1 set of valves for the scavenge pump, where lift valves are used.

1 set of valves for the lubricating oil pump.

1 bucket and rod for the lubricating oil pump.

A quantity of assorted bolts and nuts, including one set of cylinder cover studs and nuts.

Lengths of pipes suitable for the fuel delivery and the blast pipes to the cylinders, and the air delivery from the compressors to the receivers, with unions and flanges suitable for each.

PERIODICAL SURVEYS

Section 7.—(1) The engines are to be submitted to survey annually, and in addition are to be submitted to a Special Survey upon the occasion of the vessels undergoing the Special Periodical Surveys Nos. 1, 2, and 3 prescribed in the Rules, unless the machinery has been specially surveyed within a period of twelve months, in which case the Annual Survey will be sufficient. The boilers, if fitted, are to be subjected to the same surveys as required by Section 37 of the Rules for Engines and Boilers of Steam Vessels.

2. Special Surveys.—At these special surveys, the main engines and the auxiliary engines are to be examined throughout, viz.:—All the cylinders, pistons, valves and valve gears, connecting rods and guides, pumps, crank, intermediate, and thrust shafts, propellers, stern bushes sea connections and their fastenings, are to be examined. The air compressors are also to be examined. The air receivers are to be cleaned and examined and, if necessary, tested, as provided for in paragraph 3 of this Section.

3. Annual Surveys.—The whole of the parts of the engines which the engineers of the vessel open up for adjustment and overhaul should be examined and reported upon. The Survey must include, for each main engine, the examination of at least 2 pistons, 2 cylinder covers and their valves, 2 connecting rods and their brasses, both top and bottom ends. 2 of the main bearings and crank shaft journals, and 1 of the tunnel bearings. If these are all satisfactory, their condition may be taken as representing that of the other similar parts.

In the auxiliary Diesel engines, a similar course must be adopted, but in this case one of each of the parts mentioned of each engine will be sufficient, if found to be satisfactory.

The valve gears of all the Diesel engines should be examined, as far as practicable, without complete dismantling.

The air receivers must be examined internally if possible, and, together with the air pipes from the compressors, must be cleaned internally by means of steam, or otherwise. If the air receivers cannot be examined internally they must be tested by hydraulic pressure to twice the working pressure at each alternate Annual Survey, attention being specially given to the welding of the ends and of the longitudinal joints.

The pumps and air compressors must be examined and tried under working conditions. If found to be satisfactory, they need not be dismantled.

The manoeuvering of the engines must be tested under working conditions.

If the examination reveals any defects, the Surveyor should recommend such further opening up as he may consider to be necessary.

4. Record of Survey.—If the various parts mentioned in paragraphs 2 or 3 are all found to be in a satisfactory condition and the Surveyor finds that the machinery generally is in good order, he should recommend the vessel to have a fresh record of LMC.

5. Survey of Screw Shafts.—The screw shaft is to be examined annually and drawn at intervals as provided for in Section 37, Clause 3 (Rules for Engines and Boilers of Steam Vessels).

EXTRACTS FROM RULES OF AMERICAN BUREAU OF SHIPPING

20. General.—The following requirements apply to all oil engines for propelling and auxiliary purposes. All machinery parts subject to stresses are to be of sound material and the fits and clearances in accordance with the best marine practice. The passages for cooling water and lubricating oil must be carefully cleaned of sand and scale. The main bearings and the reciprocating parts should be readily accessible and lifting eyes or gear are to be fitted in way of main bearings and cylinder covers. The nuts of main bearing and connecting-rod bolts are to be secured by split pins or other efficient means.

Hand or power turning gear shall be provided for all oil engines. The engines for propelling the vessel are to be fitted with a governor or other efficient means to prevent the speeding of the engines to more than 15 per cent above the designed number of revolutions; propelling engines over 300 B.H.P. should be direct reversible. Closed crankcase engines should have suitable provisions to prevent the accumulation of gas in the crankcase.

21. Bedplate.—The bedplate or crankcase is to be of rigid construction, oil tight, and to be provided with a sufficient number of bolts to secure the same to the ship's structure. The structural arrangement for supporting and securing the main engines are to be submitted for approval.

22. Cylinders.—Cylinders, liners, cylinder covers, pistons and other castings subject to high temperatures or pressures are to be made of the best grade of cast iron or equally satisfactory material. Castings must be free from defects affecting their strength.

Cylinders using a compression pressure of over 400 lb. per square inch are to be fitted with relief valves set to not more than 1 ⅓ times the maximum working pressure, and the valve discharge should lead beyond a point of danger to life or vessel.

23. Crankshafts.—The minimum diameter of the crankshaft is to be determined by the following formula:

$$d = a\sqrt[3]{\frac{D^2PL}{f}}$$

d = Diameter of shaft in inches.
D = Diameter of cylinder in inches.
P = Initial working pressure in lb. per square inch.
L = Fore and aft length of crank over webs plus 1 (inch).
a = Factor from table below.
S = Stroke of piston in inches.
t = Thickness of crankweb in line with axis of shaft.
w = Width of crankweb perpendicular to axis of shaft.
f = 7,500 for Grade 1 forgings, Section 40-11.
f = 8,000 for Grade 2 forgings, Section 40-11.
f = 6,500 for cast steel made in accordance with Sect. 40, Par. 8.

The value of P given by the builders must be verified by the Surveyor from indicator cards during the full power trial of the engine. A set of full power indicator cards of the main engines is to accompany the Classification Report.

Subsequent adjustments for the purpose of obtaining higher initial pressures are not permitted without the special approval of the Committee.

VALUES OF "a" FOR AIR-INJECTION DIESEL ENGINES

Number of cylinders		S/L ratios							
4 cycle	2 cycle	0.7	0.8	0.9	1.0	1.1	1.2	1.3	1.4
1–2–4	1–2	1.17	1.19	1.22	1.25	1.28	1.31	1.34	1.36
3–6	3	1.19	1.22	1.25	1.28	1.32	1.35	1.38	1.41
8	4	1.20	1.24	1.27	1.30	1.33	1.37	1.40	1.43
12	6	1.22	1.25	1.29	1.32	1.36	1.39	1.42	1.45
16	8	1.25	1.29	1.33	1.36	1.40	1.44	1.47	1.50

VALUES OF "a" FOR EXPLOSIVE-COMBUSTION ENGINES

Number of cylinders		S/L ratios							
4 cycle	2 cycle	0.7	0.8	0.9	1.0	1.1	1.2	1.3	1.4
1–2–4	1–2	1.17	1.17	1.17	1.17	1.17	1.17	1.17	1.17
3–5–6	3	1.17	1.17	1.17	1.17	1.19	1.20	1.22	1.24
8	4	1.17	1.19	1.21	1.23	1.25	1.28	1.30	1.32
10–12	5–6	1.18	1.20	1.23	1.25	1.28	1.31	1.33	1.35

NOTE.—The above constants are to be used in cases where a bearing adjoins each side of each crank and where single impulses occur at equal intervals. In cases of departure from these conditions and for designs not properly coming under the two groups above, data must be submitted for the determination of shaft dimensions.

The dimensions of the crankwebs of solid shafts are to be such that wt^2 is not less than $0.4d^3$ and w^2t not less than d^3; for built-up crankshafts it is to be not less than 0.66d and w not less than 1.9 diameters of the hole in the webs. These proportioned dimensions are based on the use of the same grade of material for both shaft and webs and should be modified in accordance with the difference in the grade of the materials.

The webs are to be shrunk or forced onto the shaft and are to be fitted with dowels or keys of ample proportions to transmit at least 60 per cent of the torque. It is strongly recommended that the radius of the fillets in way of main bearings and the crank pins of solid shafts be not less than $0.05d$. The shearing stress in the coupling bolts should not exceed the torsional stress in the shafting.

24. Intermediate Shafts.—The diameter of the intermediate shafts is to be determined by the following formula:

$$d = b\sqrt[3]{\frac{D^2PS}{f}}$$

VALUES OF "b"

Air-injection Diesel engines, number of cylinders			Explosive combustion engines, Number of cylinders		
Four-cycle	Two-cycle	b	Four-cycle	Two-cycle	b
1-2-3-4-6	1-2-3	0.97	1-2-3-4	1-2	0.87
8	4	1.00	5-6	3	0.94
12	6	1.04	8	4	1.00
16	8	1.10	10-12	5-6	1 05
	12	1 20		8	1.10

Shafting for Auxiliary or indirect-drive engines may be 5 per cent less in diameter than required by the above formulae.

25. Thrustshafts.—To be 5 per cent larger in diameter than the intermediate shafts.

26. Propeller Shafts.—To be the diameter of the thrust shafts multiplied by the value of C in the table for Propeller Shafts in Sec. 34, Par. 3. The ratio in this case being that of the diameter of the Propeller to the diameter of the thrust shaft.

27. Trial.—Before final acceptance all engines must demonstrate by a running test their ability to perform satisfactorily the work for which they are intended.

28. Recommendation.—In order to minimize operating troubles it is recommended that a system of periodical inspection, cleaning, replacement and adjustments of essential engine parts be followed by the ship's engineers.

ENGINE AUXILIARIES

29. General.—The following auxiliaries are intended to cover the minimum requirements for seagoing vessels of average power and should be increased for ships of large power but may be modified for River and Harbor Boats and for sailing vessels fitted with auxiliary engines. Propelling engines of a type and design not requiring certain auxiliaries noted below are to be provided with the necessary auxiliaries or connections in duplicate. (See also par. 13.)

Pressure gauges, thermometers and relief valves are to be fitted where required.

30. Fuel Oil Transfer Pumps.—One attached pump for each engine and one independent pump, or two independent pumps.

31. Compressors for Injection Air. *Single Screw.*—One attached or one independent compressor of maximum capacity, and one independent compressor of 66 per cent capacity.

Twin Screw.—(*a*) One attached or one independent compressor per engine of maximum capacity, and one independent compressor of 66 per cent capacity for one engine.

(*b*) One independent compressor of maximum capacity for both engines and one compressor of 66 per cent capacity for both engines.

(*c*) One attached compressor for each engine, capable and arranged to supply both engines for maximum requirements and one starting air compressor of adequate size.

NOTE.—The capacities noted above refer to the maximum requirements at any speed.

32. Emergency Air Compressors.—One power operated compressor not requiring air for starting; capacity depending on size of propelling units.

33. Scavenging Pumps.—One or more attached pumps for each engine or one independent pump. Crankpit scavenging will be approved for explosive combustion engines.

34. Main Lubricating Oil Pumps.—One attached pump for each engine and one independent pump, or two independent pumps; each of a capacity for maximum requirements. Not required in cases of general multi-feed lubrication.

35. Water Cooling Pumps.—One attached pump for each engine and one independent pump, or two independent pumps; each of a capacity for maximum requirements.

36. Note.—Independent pumps denote pumps driven independently of the propelling engines. Attached pumps denote pumps driven by the propelling engines.

ENGINE PIPING

37. Fuel Oil Transfer System.—The piping arrangements for the carriage of fuel oil are to be in accordance with the requirements of Section 31.

The service tanks are to be located sufficiently high to permit gravity flow to the service pump suctions and shall be fitted with air vents leading to the atmosphere, drain cocks and a well-protected oil gauge with valves on both ends. Tanks are to be placed in drip pans provided with a drain to the oil drain well.

The pumps used for the transfer of oil are not to be used for bilge and ballast purposes; the suction pipe is to be fitted with an efficient strainer.

38. Fuel Oil Injection System.—The suction to the fuel oil injection pumps is to be fitted with a Duplex Strainer and cut-out valves are to be located at the service tanks operated from the engine room floor. The piping in the discharge line should be of seamless drawn steel and the fittings of extra heavy steel; means should be provided to discontinue the fuel supply to each individual cylinder. The joints in the pipe lines should be metal to metal or have metal gaskets; ample provision is to be made for expansion. Individual injection pumps for each cylinder are recommended. Overflows and drip pans are to have drain pipes leading to the oil drain well.

39. Starting Arrangement.—The efficiency and the capacity of the starting arrangement as required in Par. 49 is to be demonstrated to the Surveyor in attendance.

On vessels fitted with Surface Ignition Engines all reasonable safeguards must be provided to prevent fires.

40. Scavenging System.—The air supply is to be adequate for all speeds and the flow of the air should be regulated to insure uniform supply. The suction and discharge valves of scavenging pumps shall be readily accessible for examination and repair. Provision should be made to take care of excessive pressure by means of relief valves or breaking plates.

41. Injection Air System.—The discharge lines of each stage of the air compressors are to be fitted with air coolers and relief valves of ample proportions. The temperature of the air discharge from each cooler should not exceed 150 deg.

A stop valve is to be fitted in the branch line to each cylinder so located that the engine may be operated with one or more cylinders cut off. The pipe lines may be made either of seamless steel or seamless copper and should be fitted with drains.

The installation of oil separators is strongly recommended.

42. Lubricating Oil System.—The lubrication of main bearings, crankpins, crosshead or wrist pins should be by means of pump pressure or by gravity. The pistons of the engines, compressors and scavenging pumps should be lubricated by individual pumps, adjustable to the particular requirements.

Extreme care must be taken to prevent the contamination of the oil supply by sea water, escaping oil gases or carbon. The oil drawn from the crank pit or sump must run through a duplex strainer, readily accessible for cleaning. The installation of oil filters or separators is recommended.

The tanks for the storage of lubricating oil must not form part of the ship's structure.

The flashpoint of the lubricating oil should in no case be below 350 deg. F. In order to insure the continued safe operation of the engines, only the best quality of lubricants should be used.

43. Cooling-water System.—The cooling-water pumps are to have at least one high and one low sea suction; the suction line is to be fitted with duplex strainers. For any emergency connection from the fire main the cooling-water shoud be led through the duplex strainer. Means should be provided

ENGINE AUXILIARIES

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Twin Screw.—(*a*) One attached or one independent compressor per engine of maximum capacity, and one independent compressor of 66 per cent capacity for one engine.

(*b*) One independent compressor of maximum capacity for both engines and one compressor of 66 per cent capacity for both engines.

(*c*) One attached compressor for each engine, capable and arranged to supply both engines for maximum requirements and one starting air compressor of adequate size.

NOTE.—The capacities noted above refer to the maximum requirements at any speed.

32. Emergency Air Compressors.—One power operated compressor not requiring air for starting; capacity depending on size of propelling units.

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35. Water Cooling Pumps.—One attached pump for each engine and one independent pump, or two independent pumps; each of a capacity for maximum requirements.

36. Note.—Independent pumps denote pumps driven independently of the propelling engines. Attached pumps denote pumps driven by the propelling engines.

ENGINE PIPING

37. Fuel Oil Transfer System.—The piping arrangements for the carriage of fuel oil are to be in accordance with the requirements of Section 31.

The service tanks are to be located sufficiently high to permit gravity flow to the service pump suctions and shall be fitted with air vents leading to the atmosphere, drain cocks and a well-protected oil gauge with valves on both ends. Tanks are to be placed in drip pans provided with a drain to the oil drain well.

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